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AN INNOVATIVE METHOD FOR HAND PROTECTION FROM EXTREME COLD USING HEAT PIPE

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13. ABSTRACT *(Maximum 200 words)*
Due to the danger of frostbite at very low ambient temperatures, there is a need to develop new handwear technology to protect the fingers at temperatures down to about $-80^{o}F$ ($-62^{o}C$). The shortcomings of the existing technology, a glove with a heating element, are need for maintenance, size and limited capacity of the current batteries. An innovative method was investigated to transfer some of the body core thermal energy to the hands. This method involves the use of heat pipe technology, which has the advantages of very high effective heat conductivity, fast response time, flexibility low mass, compact size, and ease of maintenance. **The design uses the person's elbow area as the heat source.** The heat pipe extends along the arm and terminates at the surface of the back of the hand. From a simple model of the insulated arm and hand, the required heat transfer to the hand by the heat pipe to maintain a hand temperature of $-81^{o}F$ ($27^{o}C$) with ambient temperature of $-80^{o}F$ ($-62^{o}C$) was shown to vary from 5.8 to 21 W for an insulation 'R' value from 0.741 to $0.185 m^{2\,o}C/W$. A very efficient and flexible heat pipe was developed and tested to show the feasibility of the use of heat pipe technology in the above application. Heat capacities between 1 and 5 W were measured, depending upon orientation with respect to gravity. Due to

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the necessity of transferring 1 to 5 times this amount of heat and to provide for more uniform heat distribution, these results indicate that multiple heat pipes may be required in the design of a handwear system.

PREFACE

This report is the result of an investigation conducted by Programming and Systems Mangagement, Inc. (PSM) under Contract No. DAAK60-86-C-0068 from the United States Army Natick Research, Development and Engineering Center (Natick). It describes work and results of Phase I of the contract, which began on July 1, 1986, and was completed in February, 1987. Dr. B. Bahramian, the President of PSM, initiated the project and was responsible for the overall architecture and the conceptual design and management of the project. Dr. Amir Faghri, of Wright State University, was the project leader and had responsibility for all of the technical contents in the project. Mrs. Linda Wells, Mr. Bruce Rosen and Mr. Joseph Cohen of Natick were project officers. Mr. Scott Bennet was responsible for publishing this report.

TABLE OF CONTENTS

LIST OF FIGURES

List of Figures, continued

FIGURE | Page

LIST OF TABLES

I. METHODOLOGY

A. Introduction

Protection of the hands, especially the fingers, from frostbite in cold environments is a continuing problem. The method of protection to date has primarily been well-insulated gloves, although there has been some interest in battery-operated gloves with heating elements. The shortcomings of the latter are mostly due to the size and limited capacity of available batteries. Recently, need for hand protection in environments with ambient temperatures less than -60°F (-51°C) (or presumably wind chill factors of less than -60°F) has arisen. Consequently, insulation by itself may not be the solution since thicker insulation makes the gloves cumbersome to work with and, depending upon the temperature and insulation, may not provide adequate protection for the skin surface. Consequently, innovative designs of handwear that can provide protection to temperatures below -60°F for at least two hours and be lightweight and not overly bulky are desirable.

Since the above problem is related to heat transfer, it is reasonable to explore relatively recent developments in heat transfer technology in order to arrive at a possible solution to the problem of hand protection at very low ambient temperatures. One of the most exciting developments in heat transfer over the last 20 years is the heat pipe [1].

The heat pipe, a device with very high thermal conductance, is really an improved version of the earlier thermosyphon. Briefly, the operation of the latter is as follows. A small quantity of working liquid is placed in a tube or other such container, air is evacuated from the tube and the tube sealed. Heat is applied to the lower end, which causes some of the liquid

to vaporize. The vapor then moves due to its higher pressure to the other end of the tube where it is condensed. The liquid condensate then returns to the heated end by gravity. Since the latent heat of vaporization is large, large heat transfer rates can be achieved with small temperature differences between hot and cold ends of the tube. The heat pipe is an improvement over the thermosyphon because gravity is not needed to return the condensate to the hot end. Most heat pipes achieve this with a wick, which is fixed to the inside wall of the tube wherein capillary forces return the condensate to the evaporation section. Gravity may assist this return but it is not required. A complete source of heat pipe theory and design is given in Dunn and Reay [2].

The physiology of human exposure to extreme cold is interesting while at the same time not fully understood. Cold directly constricts the skin blood vessels such that blood flow to the hand reaches a minimum value when the local skin temperature is about 59°F (15°C) [3]. If the skin temperature is lowered below 50°F (10°C), the vasoconstriction in the fingers, hands, and other parts of the body is mixed with periods of vasodilation (widening of the skin blood vessels). This mechanism of cold-induced vasodilation is not fully understood [3] but helps to prevent frostbite by delivering warm blood to the affected areas of the skin. In lower animals whose natural habitats are cold climates, this response is well-developed, but in man it is comparitively poorly developed [4]. Consequently, a need exists for local protection of the hand and fingers from extreme cold.

The method of protection proposed here involves applying heat pipe technology directly to items for humans. As far as the authors are aware, this would be the first application of heat pipes to heat transfer for humans. As shown in the method section, the heat pipe is an extremely

efficient and fast responding method of transferring heat.

B. Significance to the U.S. Army

Developing handwear systems that afford protection from frostbite in ambient temperatures of less than -60°F (-51°C) while still allowing comfort, dexterity, and continuous use for at least two hours appears to be of considerable interest to the Army. The proposed design would provide heat transfer from warmer body surfaces to the hands and fingers down to temperatures of -80°F (-62°C). As shown in the methods, the low-end temperature for which such heat transfer can be achieved is a function of the particular working fluid in the heat pipe. This design is innovative in the sense that the problem is not being solved by providing better insulation but rather by "temperature flattening", i.e., reducing differences in temperature between unevenly heated surface areas of the human body.

There are several reasons why the heat pipe is a superior heat transfer system. These include:

1. Good thermal response time. Vapor in the heat pipe travels at nearly the speed of sound. For example, if a heat pipe 30cm long and 16mm in diameter is dipped in 176°F (80°C) water, the other end of the heat pipe will reach 80°C in only 20 seconds. As a comparison, a copper bar of the same size and under the same experimental conditions will take 3 hours for the opposite end to reach only 86°F (30°C). This characteristic is especially important for the proposed design. Rapid heat response is important to ensure that the heat protection system will be operating in a relatively short period after donning.

2. Superhigh thermal conductivity. The heat pipe has an effective thermal conductivity which is 200 times that of copper. This means that for the same size and heat transfer rate, the temperature difference required to maintain that rate would be 200 times greater for the copper bar than the heat pipe. In terms of the proposed design, this means that a 200 times greater heat transfer rate can be obtained with a heat pipe than with the same size copper rod for the same temperature difference between the ends. Since temperature differences on the body surface are not large (typically, the temperature difference between the armpit and the hand surface in a cold environment is about 16°F (9°C) [5]), the high thermal conductivity of a heat pipe is necessary to achieve desired heat transfer rates.

3. Temperature uniformity. Uniform temperatures of each part in the heat pipe can be obtained throughout its length. What this means for the design is that a temperature gradient is produced along the length of the heat pipe, but that at each section the temperature is constant at all points around the heat pipe. Thus, the heat pipe system interfaced between the armpit and the hand running along the arm's length would simulate a normal temperature distribution along the arm and hand, even though they are in cold environments.

4. Flexibility in design. The heat pipe allows heat transfer between the two ends and the actual configuration used does not restrict the ability to accomplish the transfer. Thus the materials used to construct the casing may be flexible and easily conform to the individual wearing the device, allowing as much freedom of movement as is necessary and can be practically achieved. The working fluid

inside the heat pipe becomes the one thing that must be selected for the operating range of temperatures involved. At the present time, using ammonia as the working fluid inside the completely sealed heat pipe seems most appropriate, mainly because its useful operating range extends down to -80°F (-62°C) [2].

5. Ease of Maintenance. Heat pipes function without being driven electrically or mechanically. Thermal and chemical stability on the inside is adequate. Perfect operating conditions are maintained over long periods of time without maintenance. This is an especially desirable feature for ensuring continued use of the device by personnel.

6. Lightweight and compact. While this advantage is not important in many engineering applications, it is especially important when interfacing a heat pipe to an *individual* who must move and work comfortably while wearing the system. Even if heat pipes had all the superior characteristics listed above, if this last one were lacking, the hand protection system design using heat pipes would not be feasible.

C. Proposed Method

The hand protection system proposed here intends to use heat pipe technology to transfer heat to the hand from surface areas of the human body that stay relatively warm. The hands, particularly the fingers, are areas susceptible to frostbite in cold environments.

The heat pipe basically consists of a sealed container which is evacuated and filled with a working fluid. Usually a wick, which may be made out of a variety of materials, is adjacent to the inner wall of the

container. One end absorbs the heat to be transferred and is called the evaporation section. The working fluid evaporates due to this heat and flows toward the other end due to its higher pressure at nearly sonic speed. The latter end is the condenser section, which then gives up the heat of the condensing vapor. The capillary force of the wick then returns the condensate back to the evaporator section to finish the cycle. This process continues as long as a temperature difference exists between the ends.

PSM's design will use the elbow area of the person as the heat source located in close proximity to the pipes' evaporator section. The heat pipes then run down the arm and terminate at the surface of the back of the hand, which is where the condensation occurs. The development of the exact cross-sectional shape, overall configuration, and materials of the container was one of the primary goals of the design in the first phase of this study.

D. Phase I Objectives

The following objectives were completed during the phase I of this study:

1. Physiological modeling and analysis of heat loss from hands and fingers. This step was necessary in order to know what heat rate must be transferred by the heat pipe system.
2. Analysis of the heat pipe capacity. Knowledge of 1. allows calculations concerning the size of the system.
3. Material selection. Flexibility is a necessary feature of the system. In the search for the right materials and the system's construction, complete freedom of movement of the arms is an absolute must.

4. Wick design. A variety of wick designs have been used in other applications. The key consideration here is the matching of wick flexibility to container flexibility while at the same time supplying the necessary capillary force.

5. Overall configuration design. This part of the design considers the actual three-dimensional configuration of the heat pipes from the elbows to the fingers. Again, flexibility and comfort are of utmost importance.

6. Building an experimental heat pipe model.

7. Actual observation of an experimental model and testing.

E. Summary

Due to the danger of finger frostbite at very low ambient temperatures, there is a need to develop handwear for protection at temperatures lower than that provided by currently available handwear, which is to about -60°F (-51°C). Since the human physiologic blood flow response to lowered skin temperature is inadequate to provide the heating necessary to protect the hands and fingers, we propose an innovative method to transfer some of the body core thermal energy to the hands. This method involves the use of the heat pipe, a new technology which has very high effective heat conductivity, fast response time, flexibility, low weight, and a compact form. With the proper working fluid inside the heat pipe, operating temperatures can be extended to -80°F (-62°C). The emphasis in this phase of the research is to propose and design a working model of the handwear system that would allow the user to wear the system for a minimum of two hours.

II. MODELING HEAT LOSS FROM HANDS IN A COLD ENVIRONMENT

A. Introduction

In moderately cool environments, i.e., between 32°F (0°C) and room temperature, the body attempts to conserve energy by automatic control of the channels for venous return of blood, especially in the extremities. This is shown in Fig. 1, which is a schematic diagram of the arm and hand. When the body wishes to reject heat, blood flow through the superficial veins near the skin surface is increased, i.e., the valves in Fig. 1 representing the arterial-venous anastomoses are influenced to open, or vasodilation occurs in these vessels. When conservation of body heat is vital, the valves shut or vasoconstriction occurs and blood is bypassed through deeper veins in proximity to the arteries. These automatic mechanisms raise or lower the temperature gradient for heat transfer by conduction in the subskin layers, which have no blood vessels [5]. Tissues without blood flow are poor conductors of heat, therefore control of blood flow is extremely important in the control of heat transfer between the body's core and the skin. In cool environments, significant countercurrent heat exchange occurs between major arteries and close overlying veins as shown in Fig. 1. The precooling of arterial blood by heat loss to adjacent venous blood reduces limb temperature and thus heat loss to the environment.

The mechanism described above provides a means to conserve heat, which is desirable for cool ambient temperatures above freezing. However, below freezing, protecting the tissue from freezing becomes of paramount importance. Consequently, under subfreezing conditions, blood flow through superficial veins actually does occur and raises temperature levels above

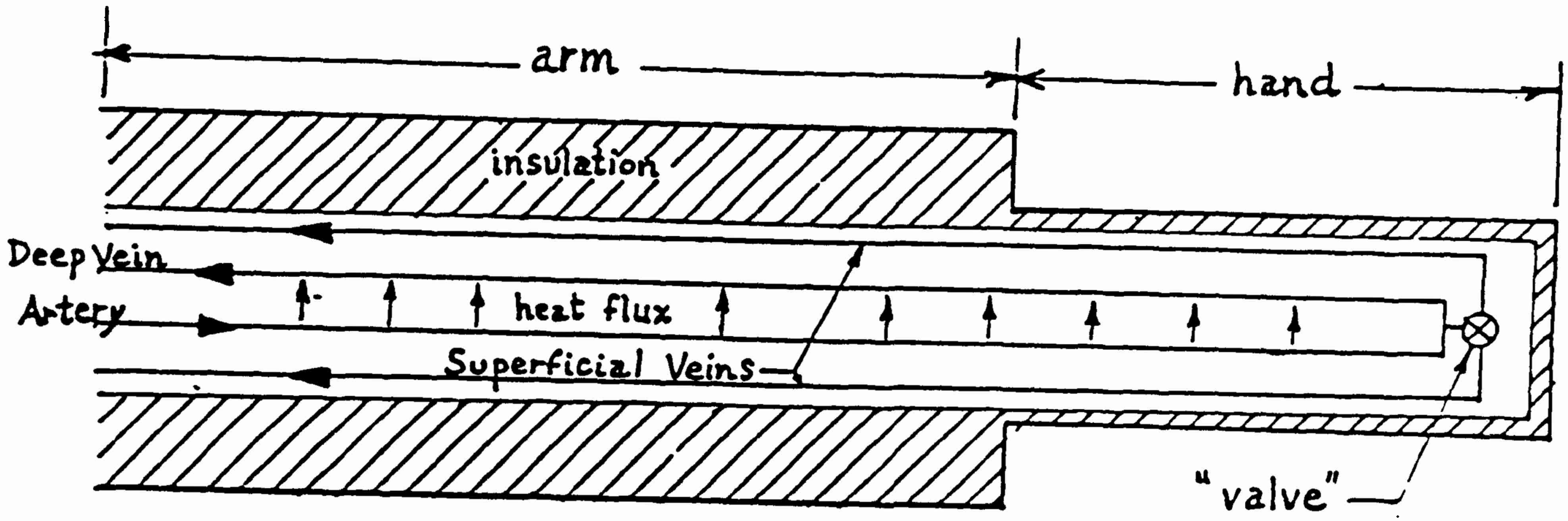


Figure 1. Schematic Drawing of Hand and Arm Model for Heat Transfer. (Adapted from Refs 5, 8)

freezing [6]. The latter is called "paradoxical vasodilation" and is observed primarily in the extremities, such as the fingers [6]. Once the temperature of the affected extremity increases due to the blood flow, subsequent vasoconstriction causes temperature reduction again and cyclical variation in temperature [6]. Paradoxical vasodilation is poorly developed in people from nonfrigid climates but is better developed in natives of frigid climates, such as Eskimos [6]. Some degree of acclimation to cold in the fingers and hands of people regularly exposed to low temperatures (fishermen, polar explorers, etc.) has been reported, apparently due to less initial response of the vasoconstriction (heat conserving) mechanism and more rapid onset of the cold vasodilation (tissue protecting) mechanism [6].

Even a well-developed internal mechanism for protecting tissues of the extremities from freezing must have its limits. Since ambient temperatures and "wind chill" exceed these limits in certain times and locations, there is considerable interest in developing improved cold protection systems for the hands, which are one of the first extremities to suffer damage from extreme cold. PSM has proposed that heat pipe technology be used to achieve a more uniform temperature distribution along the surface of the arm, similar to the result of paradoxical vasodilation. It is hoped this method will allow the higher temperatures on the mid and upper arms to provide the driving temperature difference for heat transfer to the hand and fingers via a suitably designed heat pipe system. The higher temperatures on the mid and upper arms appear to be maintained by some combination of increased metabolism, local blood flow and sufficient insulation [6]. Veghte [7] has shown that for nude subjects exposed to 39°F (4°C) air, steady-state temperature differences of up to 40°F (4.5°C)

exist between various regions of the arm and the hand. Even in a comfortable environment of 73°F (23°C), Veghte measured differences of up to 35°F (1.7°C). Thus, one would expect that a well-insulated person in an extremely cold environment would have skin temperatures somewhere between Veghte's comfortable and cold nude person, with arm to hand temperature differences of 35°F (2°C) to 39°F (4°C). If the hand and fingers were initially even colder, then driving temperatures greater than the latter would exist. Of course, this information is vital in the heat pipe design.

One of the first questions to be answered in designing a heat pipe system to "flatten" the axial temperature distribution down the arm is what heat transfer rate must be delivered to the hand to maintain it at some desired temperature. Lacking experimental data on heat loss from insulation-protected hands, we will develop a heat transfer model of the hand and its covering insulation.

B. Heat Transfer Model of the Hand and Covering

The model of the arm and hand shown in Fig. 1 is our starting point. Considering the hand and its insulation (hereafter, the system) in Fig. 1 we make the basic assumptions (assumptions are numbered throughout the text):

Assumption 1

The system is modeled by a simple geometric shape, e.g., a flat plate or cylinder.

Assumption 2

Transient effects are not considered, i.e., steady-state heat

transfer.

A heat balance on the system yields

$$\dot{E}_{in} + \dot{E}_{gen} - \dot{E}_{out} = \dot{E}_{stored} = 0 \qquad (1)$$

where $\dot{E}_{stored} = 0$ due to Assumption 2. Since the system has little metabolic activity [8], metabolic heat generation can be neglected, thus

Assumption 3

$\dot{E}_{gen} = 0.$

Energy input to the system results from net energy carried in by blood flow, axial conduction of heat through the tissues, axial conduction of heat via the heat pipe, radiation absorbed from the environment, and radiant energy absorbed from the sun. Energy leaving the system results from ‹ ‹vection and radiation from the surface of the system. We are neglecting heat loss by diffusion of water through the skin and insulation. In symbolic form, Eq. (1) becomes

$$\dot{m}_{blood} C_p (T_a - T_v) + Q_{cond} + Q_{htpipe} + \alpha_{sky} A G_{sky} + \alpha_{sky} A G_{sun} -$$

$$Q_{conv} - E_{rad} = 0 \qquad (2)$$

where

$\dot{m}_{blood}$ = mass flow of blood in the arm

C_p = heat capacity of blood

T_a = arterial temperature of the blood at the wrist

T_v = venous temperature of the blood at the wrist

G_{sky} = earth irradiation due to atmospheric emission

α_{sky} = surface absorptivity of atmospheric emission

A = surface area for heat transfer

G_{sun} = solar irradiation

α_{sun} = solar absorptivity of the surface

Q. = heat transfer mode as specified by subscript.

Making

Assumption 4

Q_{cond} at wrist = 0

and inserting the following relations for the heat transfer modes:

$$G_{sky} = \sigma T_{sky}^4 \tag{3}$$

$$\dot{E}_{rad} = \epsilon A\sigma T^4 \tag{4}$$

$$Q_{conv} = hA(T_s - T_\infty) \tag{5}$$

where

T_{sky} = effective temperature of the atmosphere

T_s = surface temperature

T_∞ = ambient temperature

σ = Stefan-Boltzmann constant (5.67 x 10^{-8} W/m^2K^4)

ϵ = Surface emissivity

h = convective heat transfer coefficient,

due to the similarity in the spectrum of G_{sky} and E_{rad} (see reference 9) $\alpha_{sky} = \epsilon$ and then Eq. (2) becomes

$$\dot{m}_{blood}C_p(T_a-T_v) + Q_{htpipe} + \epsilon\sigma A(T_{sky}^4-T_s^4) + \alpha_{sun}AG_{sun} - hA(T_s-T_\infty) = 0 \tag{6}$$

A steady-state energy balance on the insulation surrounding the hand yields

$$\frac{k_g A}{\delta}(T_h - T_s) + \alpha_{sun}AG_{sun} - \epsilon\sigma A(T_s^4 - T_{sky}^4) - hA(T_s - T_\infty) = 0 \tag{7}$$

where T_h is an average hand temperature, which is specified, and k_g is the effective thermal conductivity of the gloves. T_s can be computed from (7) for known conditions and combining (6) and (7) results in

$$Q_{heatpipe} = \frac{k_g A}{\delta}(T_h - T_s) - \dot{m}_{blood}C_p(T_a - T_v). \tag{8}$$

C. Convection Heat Transfer from Glove Surface

1. Free Convection

Other relations must be used to obtain h, since it depends on environmental conditions. For example, let us assume that a hand is to be modeled as a flat plate with a surface area A = 0.05 m^2 [8]. Assuming a horizontal plate, the heat transfer coefficient from the upper surface of a heated plate in natural or free convection is given by

$$Nu = \frac{hL}{k} = 0.54\ (GrPr)^{1/4} \quad \text{for } 10^4 < GrPr < 10^7 \tag{9}$$

where L is a characteristic length given by

$$L = A/P \tag{10}$$

where A and P is the surface area and perimeter of the plate, respectively, and k is the thermal conductivity of the surrounding air.

Using representative values for a moderate-sized glove, $L = 11.6 \times 21.6\ cm^2/2\ (11.6 + 21.6) = 3.8$ cm. This is an estimate based on the gloved hand being an 11.6 cm x 21.6 cm rectangular plate, yielding a total surface area (both sides) of $A = 0.05\ m^2$. Eq. (9) and (10) are given in [4]. Gr and Pr are the Grashof and Prandtl numbers, respectively, given by

$$Gr = L^3 g\beta\ (T_s - T_\infty)/\nu^2 \tag{11}$$

$$Pr = C_p\ \mu/K. \tag{12}$$

All properties in the above relations are for air evaluated at the film temperature given by $T_f = (T_s + T_\infty)/2$. A preliminary calculation of T_f indicated that T_f = 225°K (-48°C), where T_∞ = 211°K (-62°C). Thus h for the upper surface can be evaluated in terms of $\Delta T = T_s - T_\infty$ and L as

$$h_{up} = 1.493\ (\Delta T/L)^{1/4} \tag{13}$$

where h_{up} has units of W/m^2K, ΔT is in K and L is in m. The h of the lower surface of a heated plate is half that of the upper surface, so

$$h_{low} = 0.7465\ (\Delta T/L)^{1/4} \tag{14}$$

thus an average h for a horizontal plate heated on both sides in calm air at T_f = 225K is

$$h_{ave} = 1.12\ (\Delta T/L)^{1/4}\ . \quad (15)$$

Inserting L = 0.038m yields

$$h_{ave} = 2.54\ \Delta T^{1/4} = C\Delta T^{1/4}\ . \quad (16)$$

Calculations also show that the product of Gr and Pr are in the range of equation validity. A vertical plate has an identical form of relation for h and calculations show similar value for C in (16). Consequently, we will take this value as being representative of free convection from the surface. Note that the convection coefficient is mildly dependent (1/4 power) on the temperature difference, unlike forced convection, which is not. This value of C is about 23% higher than that reported by Nielsen and Pedersen [10] for free convection from nude persons in air. The larger values we have computed are primarily due to the very low environmental temperature involved and its effect on air properties. Our values also average about 20% higher than the simplified values in air suggested by Holman for heated horizontal plates [11].

2. Forced Convection

PSM studied flat plate models in different orientations with respect to the air flow and cylindrical models in cross flow to estimate forced convection coefficients. Comparisons to experimental results in humans or mannikins are made as well. The form for most correlations in forced convection is

$$Nu = hL/k = CRe^{m}Pr^{n} \quad (17)$$

where values of C, m, and n depend on geometry and flow conditions. Re is the Reynolds number of the flow. In most instances properties are evaluated at T_f, but in more recent work, evaluation at T_∞ and including a correction factor has been reported [9]. When the flow is parallel to the plate and laminar, m = 1/2, n = 1/3 and C = 0.664, where L is the length of the plate [9]. Taking L = 0.116 m (hand width) and evaluating air properties at the aforementioned estimate of T_f = 225 K, yields

$$h_{\text{plate parallel flow}} = 11.5\ U_\infty^{1/2} \tag{18}$$

where U_∞ is the free stream air velocity.

This is remarkably similar to reported values for forced convection from nude persons in parallel air flow (summarized in [5]). A vertical plate in cross flow [9] has C = 0.228, m = 0.731, and n = 1/3, which yields

$$h_{\text{plate cross flow}} = 34.7\ U_\infty^{0.731} \tag{19}$$

This is considerably larger than for nude persons in cross flow, which indicates that whole body heat transfer in cross flow is probably not well modeled by a plate. However, lacking evidence for regional heat transfer from parts of the body, this relation may apply to the gloved hand in cross flow.

3. Circular Cylinder in Cross Flow

With a mitten on, the hand may present more of a cylindrical geometry. Zhukauskas [12] suggests that C = 0.26, m = 0.6 and n = 1/3 in Eq. (17) for $10^3 < Re_D < 2 \times 10^5$ where the characteristic length L is the cylinder

diameter D and properties are evaluated at the ambient temperature, T_∞. A correction factor $(Pr_\infty/Pr_s)^{1/4}$ is also a multiplicative factor, but for the temperature differences involved here, this is nearly unity. Evaluating properties at $T_\infty = 211$ K and letting $D = 0.116$ m yields

$$h_{\text{cylinder cross flow}} = 11.6\ U_\infty^{0.6} \tag{20}$$

This is also in fair agreement with several studies measuring coefficients for nude persons in cross flow [5]. Most of the difference between the h value given here and those reported for cross flow in human subjects can be attributed to the kinematic viscosity being smaller by a factor of about 1.67 at $T_f = 225$ K than at room temperature. This results in a 36% increase in the constant term in (20) over that which would be predicted for room temperature. After an adjustment for the temperature dependence of gas properties, the above result is within 30% of experimental data for persons in cross flow. This fact is a remarkable similarity, since heat transfer correlations are only accurate to about this percentage [9], and the human form is certainly more complex than a cylinder.

D. <u>Results of the Model</u>

The method of solution was to solve the nonlinear algebraic equation (7) for surface temperature T_s and then insert this value into (8) to compute the required heat transfer of the heat pipe. The following assumptions were made, resulting in a high estimate of $Q_{heatpipe}$.

<u>Assumption 5</u>

$G_{sun} = 0$

Assumption 6

$\dot{m}_{blood} = 0$. The vasoconstriction mechanism if operative reduces the blood flow to the hand anyway and any blood flow coupled with a positive $T_a - T_v$ results in a reduction in the transfer required of the heat pipe. Lih [8] has estimated that an input of 5.5 W may be contributed by the blood flow in moderately cool environments. However, in extreme cold environments, blood flow rate may be much below Lih's estimate, thus we will assume a zero contribution. Certainly it would not result in a net removal of heat.

Assumption 7

$T_{sky} = 230°K$. This is an estimate from Incropera and DeWitt [9] for the extreme case of a cold clear sky.

The solution to (7) was obtained for several values of T_h, h, ϵ, and k_g/δ values. These are now summarized.

1. Free Convection

Equation (16) was used for h. Hand temperatures T_h of 273 K (0°C), 286 K (13°C) and 300 K (27°C) were specified. Three values of k_g/δ, namely, 1.35, 2.7, and 5.4 W/m^2K were studied. To assess the relative importance of radiation, $\epsilon = 0.05$ and 1.0 were also studied. Results of heat required to maintain the above hand temperatures are shown in Figs. 2 and 3.

Fig. 2 is a plot of the required heat versus the desired hand temperature to be maintained. Note that over the range of hand temperatures studied, heat required increased nearly linearly with hand temperature. The effect of insulation is also demonstrated, showing the

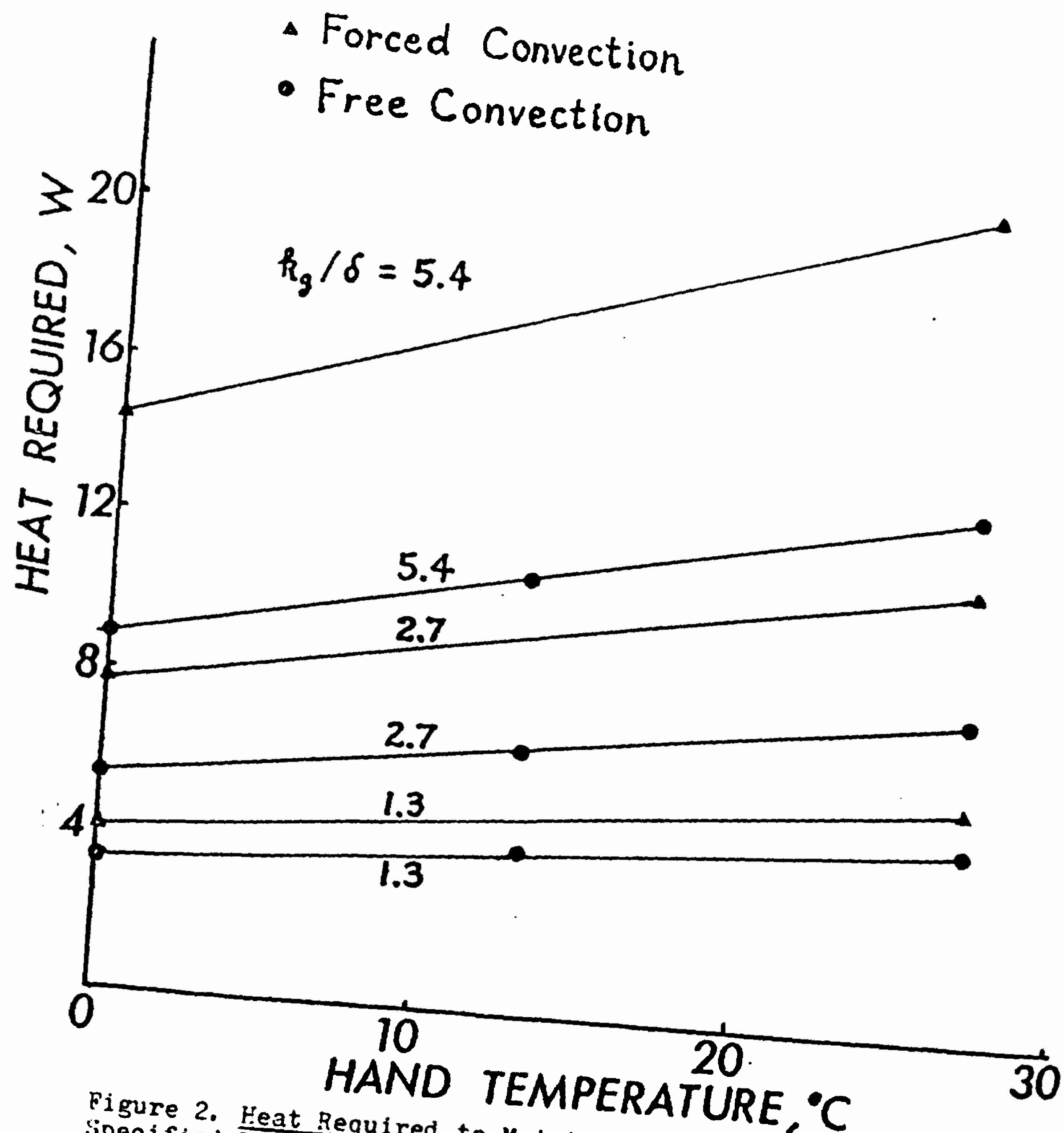


Figure 2. Heat Required to Maintain Hand Temperatures at Specified Temperatures in Free and Forced Convection Environments for Several Values of Insulation Conductivity to Thickness Ratio.

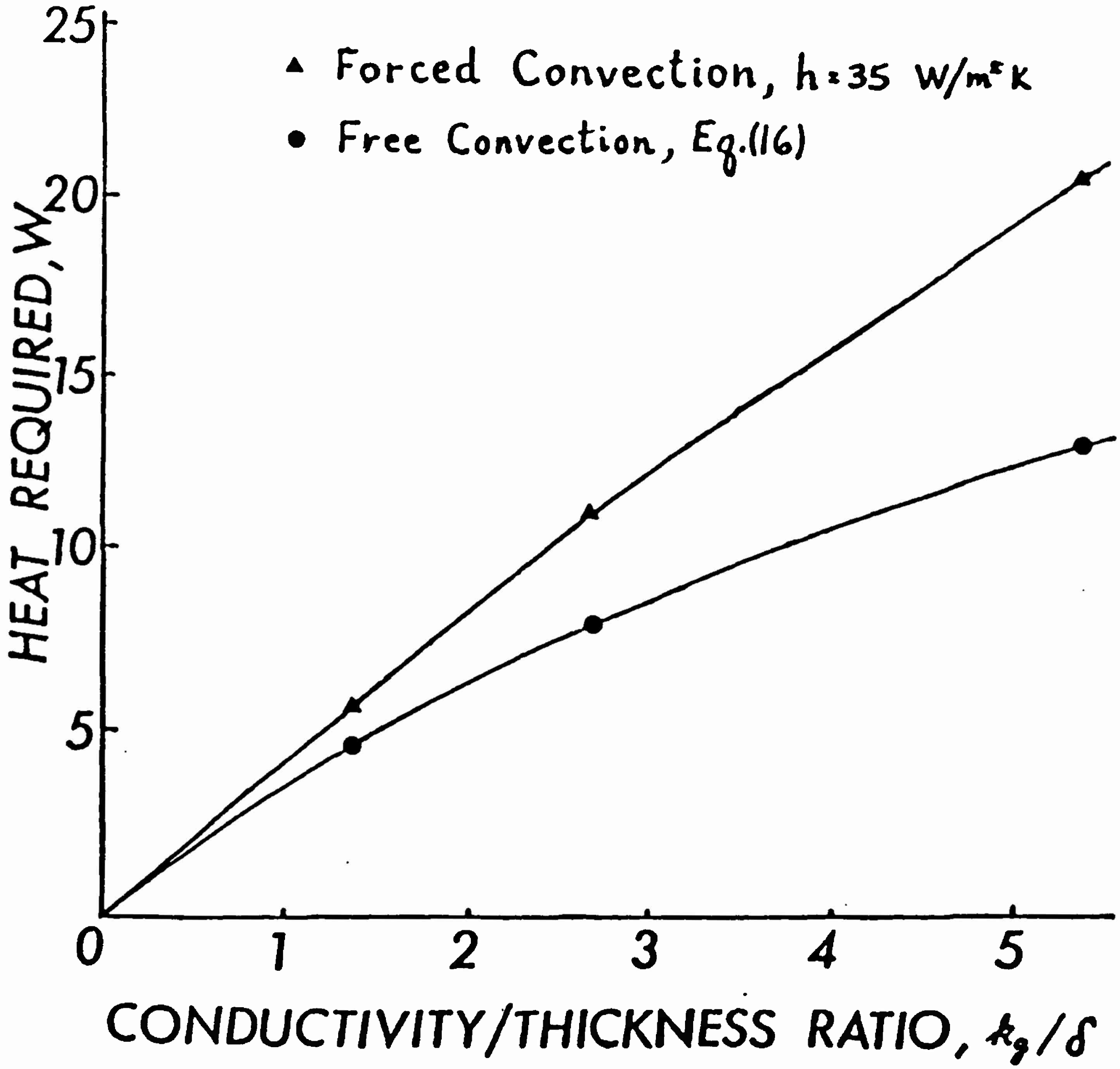


Figure 3. Heat Required to Maintain Hand Temperature at 27°C for Free and Forced Convection at 3 Values of Insulation Conductivity to Thickness Ratio.

effectiveness of smaller values of k_g/δ in reducing the amount of heat required, even when free convection is considered. For example, reducing the conductivity to thickness ratio by 50% results in a 40% decrease in the heat required at T_h = 300°K.

The effect of surface emissivity was also assessed. Values of ϵ = 0.05 and 1.0 were used to compute T_s and $Q_{heatpipe}$. For ϵ = 1.0 (blackbody), the ratio of heat radiated to heat convected was 17%, for ϵ = 0.05 the ratio was 0.5%. Consequently, at these temperatures, radiation plays a small role in the heat loss. Similar ratios in forced convection would be even lower.

2. Forced Convection

The forced convection used was that for a plate in cross flow, Eq. (19). The latter yields the highest coefficients and thus would probably give an upper bound on heat required. Specifically, the results shown in Figs. 2 and 3 are for $\bar{h}$ = 35 W/m^2K which corresponds to an air speed of 1 m/s using Eq. (19) or 6.4 m/s if Eq. (20) is used. At the lowest k_g/δ ratio studied (1.35), note that forced convection heat transfer is only slightly higher than that for free convection. However, as the effectiveness of insulation is decreased, a wider divergence between forced and free convection heat transfer is seen. As was seen for free convection, a 50% reduction in k_g/δ resulted in a nearly equal percent decrease in heat transferred.

Fig. 3 shows required heat to maintain hand temperature plotted against the conductivity to thickness ratio for the glove or mitten insulation. Results for $\bar{T}_h$ = 300 K are shown for free and forced convection. The heat required increases with k_g/δ as expected. Again a

similar amount of heat is required for both free and forced convection at the lowest kg/δ studied. This plot reinforced the neccesity of adequate insulation, not only to provide protection to the unaided hand but also to lower the requirements for supplemental heat transfer.

E. Summary

A simple model of heat transfer from the insulated hand has been made to assess heat loss in cold environments. This loss needs to be known in order to design protective wear for the hand using heat pipe technology. Heat losses by convection, both free and forced, and radiation were included. Other heat input effects, such as that due to blood flow and radiant flux of the sun, were included in the general description, but were set to zero in the solution of the energy balance equations. Thus, the estimate of the amount of heat which must be transferred to the hand by the heat pipe to maintain a given hand temperature is on the high side.

Over a hand temperature range of 0° C to 27° C (32° F to 80.6° F), the amount of heat required to maintain hand temperature increased linearly with hand temperature and depended upon insulation and convective resistances to heat transfer. For free convection at a hand temperature of -16° F, ambient temperature of -80° F, the required heat transfer varied from 4.7 to 13.2 W for insulation "R" values (i.e. δ/k_g) from 0.741 to 0.185 °C m²/W. The R value is the insulation thickness in m divided by its thermal conductivity in W/m° C. Forced convection at wind speeds producing a convection coefficient of 35.3 W/m² °C increased the required heat transfer from 5.8 to 21 W over the same range of R values above. Depending upon the geometric model used for the hand, this convection coefficient corresponds to wind velocities from 1.0 to 6.4 m/s. Radiation

heat losses were found to be relatively small in comparison with convective losses. For example, the highest percentage (17%) of radiation to convection heat losses was found when the insulation surface has the emissivity of a blackbody, i.e., unity, and the air is calm (free convection). The importance of insulation in lowering the heat losses and thus heat transfer required by the heat pipe is demonstrated. At the highest R value studied (0.741 $^\circ C\,m^2/W$) forced convection produced a heat loss only 23% higher than free convection at a hand temperature of 27°C, whereas at the lowest R value studied (0.185 $^\circ C\,m^2/W$, a four fold decrease), forced convection losses were 60% greater than for free convection.

III. PROPOSED MODEL CONFIGURATION FOR THE HEAT PIPE GLOVE

A. Analysis of Different Heat Sources

In order to determine the location of heating source zone for the flexible heat pipe, one needs to measure the skin temperatures at different locations of the body in the cold environment. Precise measurement of skin temperatures with standard thermocouple wires in direct contact with surface skin presents many problems. H. Veghte [7] used scanning infrared radiometers for measuring the temperature of 15 nude subjects exposed to environments of 39°F (4°C) and 73°F (23°C) for two hours and sweating subjects exposed to an environment of 81°F (27°C) for 10 minutes. Veghte [7] measured surface temperature of 41 different body areas by relating calculated temperature values for a gray scale on each thermogram with densitometer readings. Surface temperatures were found to be more variable in cold, 46°F (8°C) to 61°F (15°C), with an average mean temperature of 52°F (11°C). The nose, pectoral areas, patella, gluteous maximus and fatty tissue about the waist were cold regions whereas the upper chest, forehead and spinal column were warm regions. Based on this information, it appears the forehead, upper chest, back of the neck or spinal cord are the ideal locations of heat source for the evaporator section of the heat pipe. For reasons of comfort and flexibility, PSM decided to use the elbow as the evaporator source.

B. Analysis of Preliminary Models

Five different models are proposed for the general configuration of the heat pipe based on the analysis of section III-A. The first three models require flat, flexible heat pipes and the proposed configurations for these are presented in Figures 4 to 6. The differences among the first three

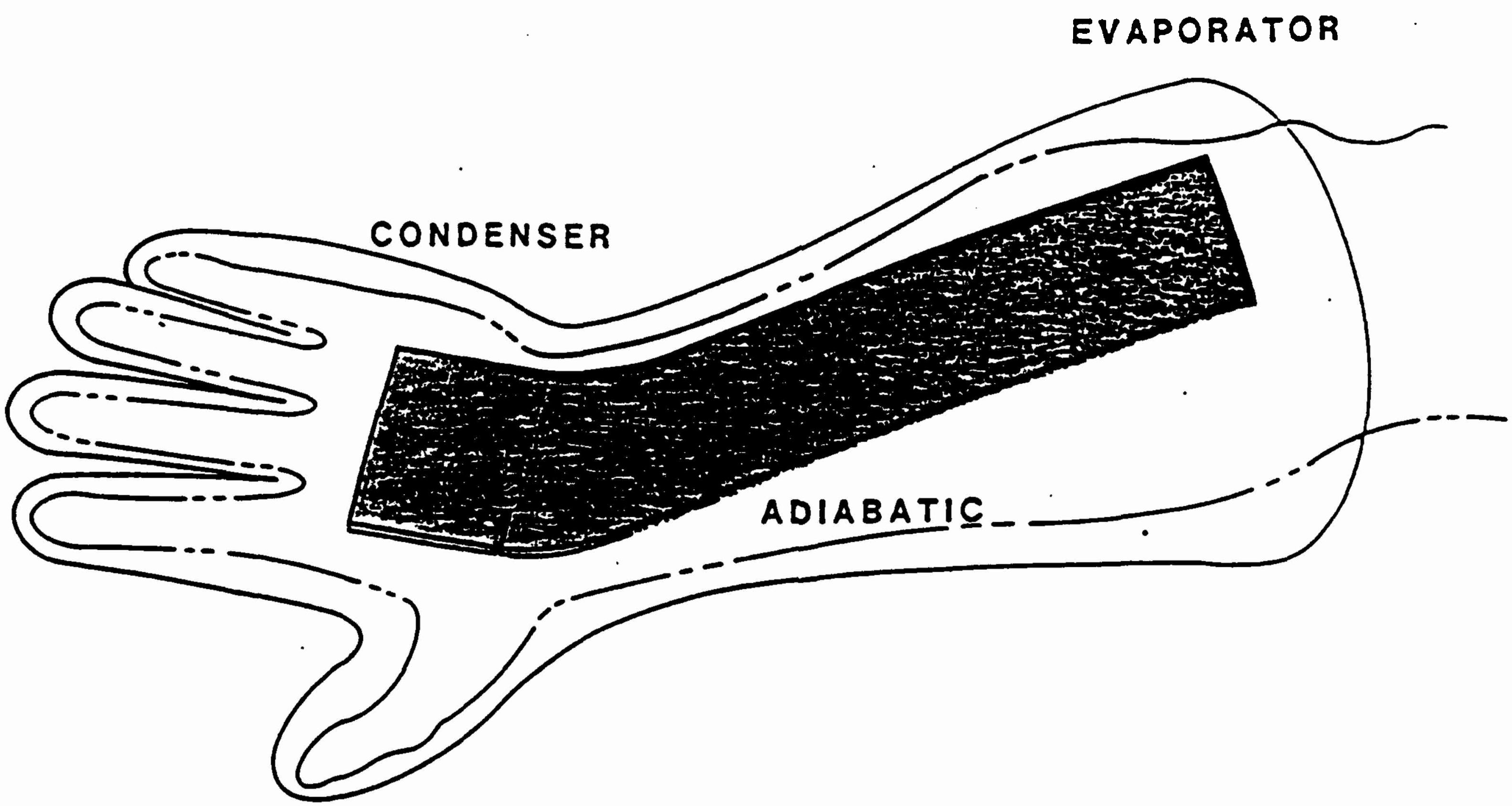


Figure 4. Hand Glove Configuration Model I

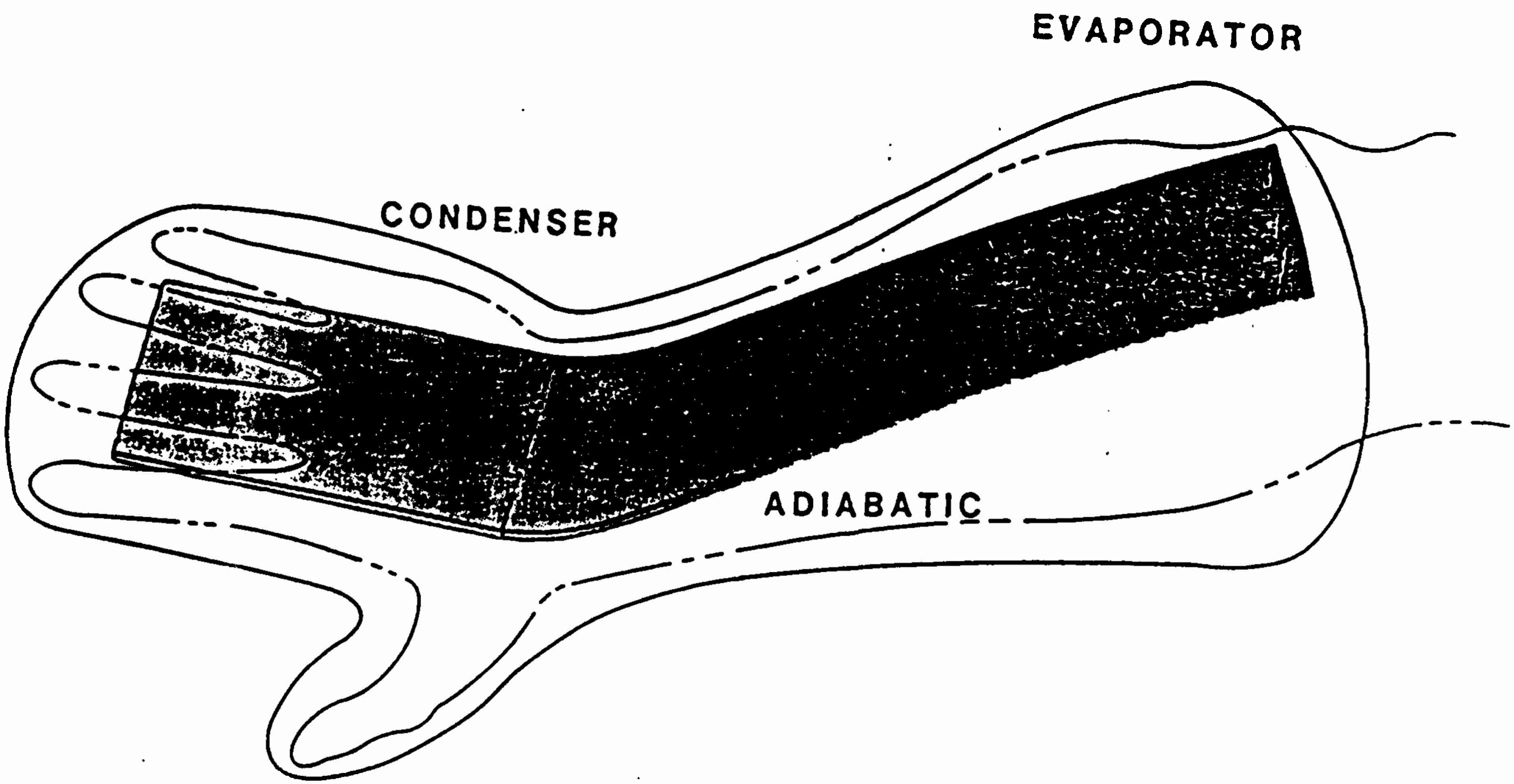


Figure 5. Hand Glove Configuration Model II

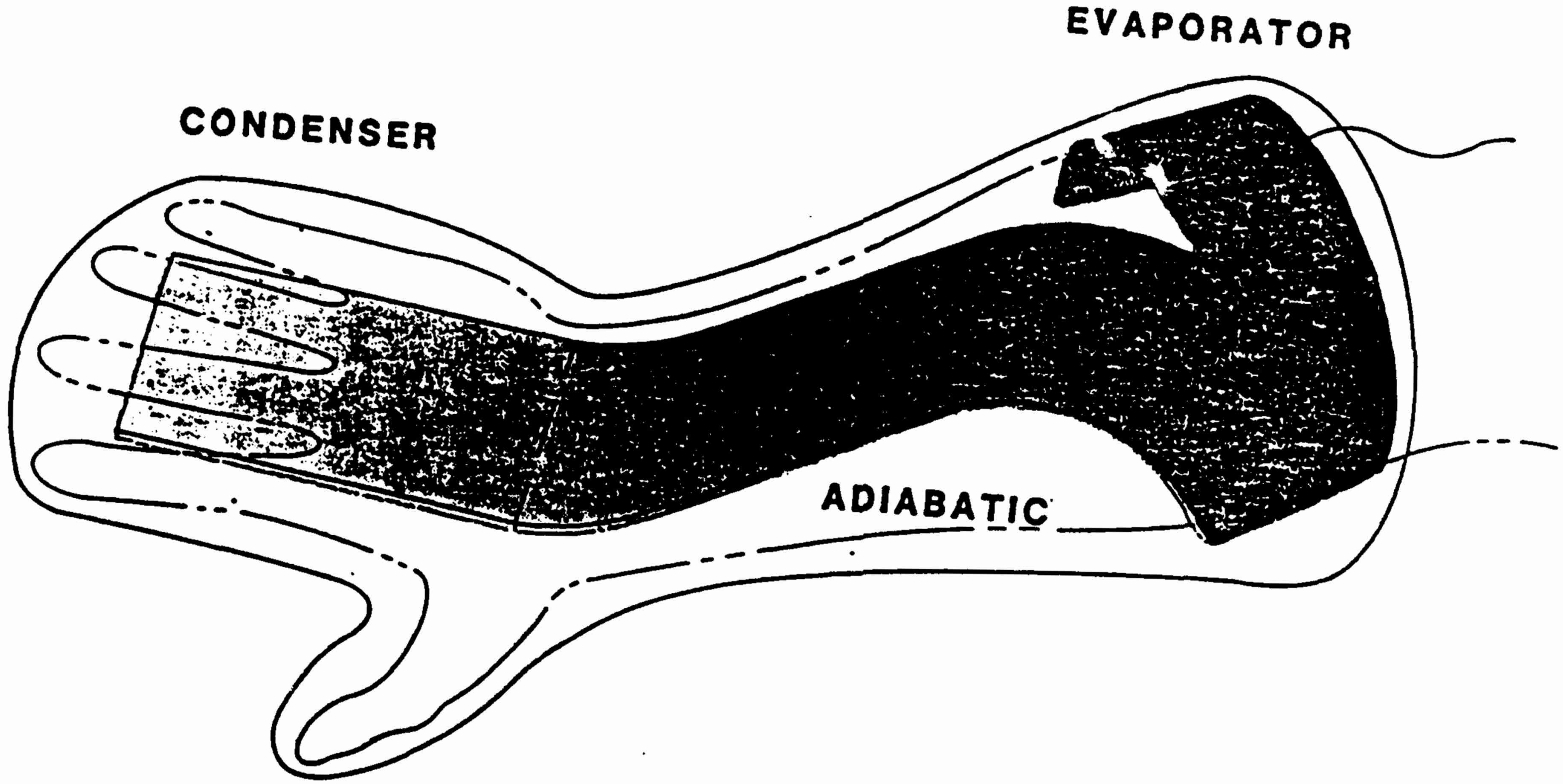


Figure 6. Hand Glove Configuration Model III

models are the locations of the evaporator and condenser section of the heat pipe. Models I & II use the posterior part of the elbow as the heating source while Model III uses the anterior part. Models II and III use the posterior part of the fingers for the primary cooling while Model I uses the posterior part of the hand for the same purpose. Models IV and V use flexible circular heat pipes and are shown in Figures 7 and 8. In both of these models, the fingers are used as the condenser zone. Model V uses the anterior part of the elbow and Model IV uses the posterior part of the elbow as the evaporator zone. Five circular flexible heat pipes are required for each hand. At the present time, Model V is selected as an ideal configuration for building a laboratory model and testing.

C. Heat Pipe Working Fluid

The three basic components of a heat pipe are:

1. The working fluid.
2. The wick or capillary structure.
3. The container.

In the selection process of a suitable combination of the above, a number of conflicting factors may arise and therefore it would be reasonable to select the working fluid first. A first consideration is the identification of a suitable working fluid in the operating vapor temperature range. For this particular application, Freon R12 would be the ideal working fluid since the melting point is -108°F (-78°C) and the boiling point at atmospheric pressure is -27°F (-33°C). The useful range for Freon R12 is from -60°C to 100°C. The other requirements are:

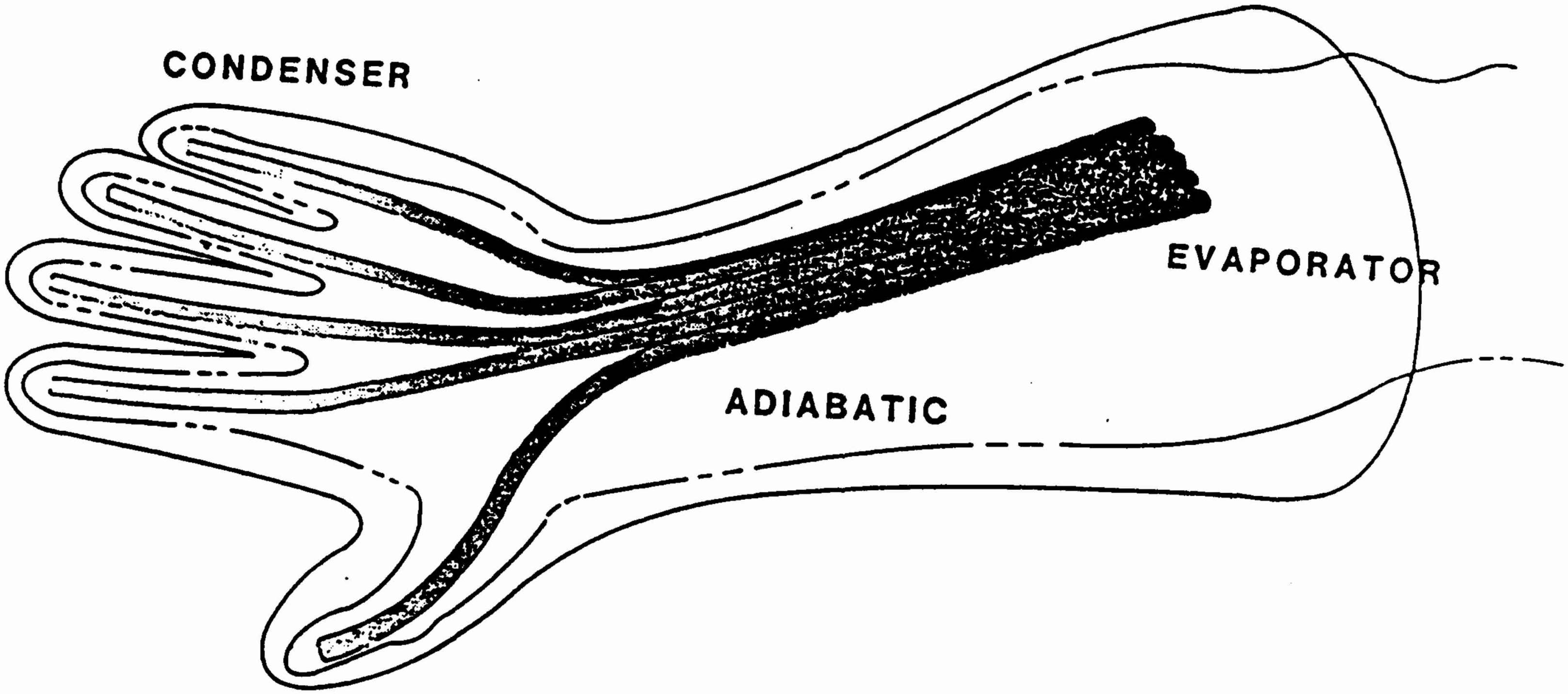


Figure 7. Hand Glove Configuration Model IV

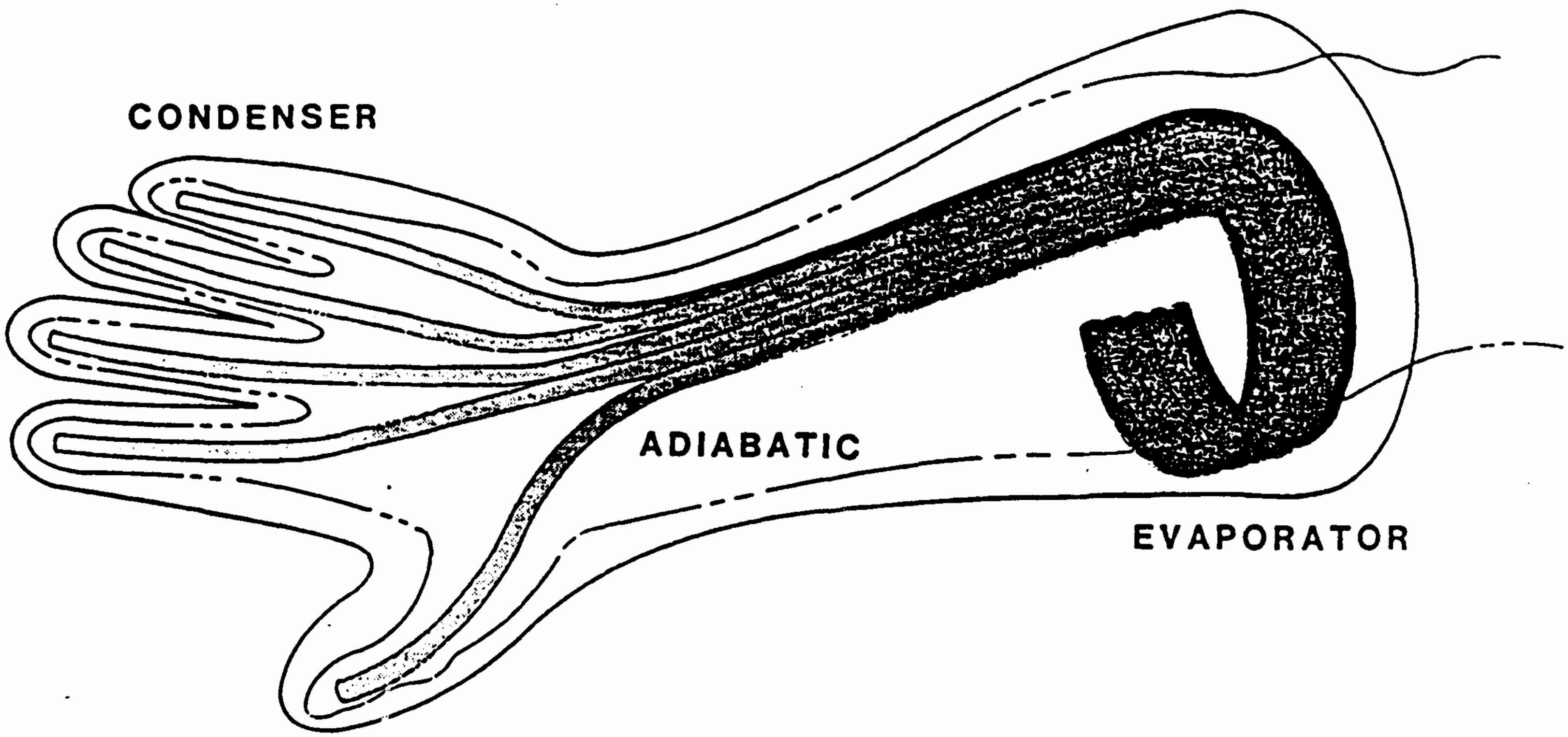


Figure 8. Hand Glove Configuration Model V

1. Compatibility with wick and wall material.
2. Wetability of wick and wall material.
3. High latent heat.
4. High thermal conductivity.
5. High surface tension.

These requirements are considered in the subsequent section for the selection of the wick and containers.

D. Heat Pipe Wick Structure

The primary purpose of the wick is to generate capillary pressure to transport the working fluid from the condenser to the evaporator. It must also be able to distribute the liquid around the evaporator section to any areas where heat is likely to be received by the heat pipe. The carbon fiber wick is capable of providing three times the capillary force of a screen mesh. The carbon fiber wicks, consisting of very fine fibers (7-10 μm), also display a high thermal conductivity and they are chemically and thermally stable, mechanically strong and lightweight.

E. Heat Pipe Container Structure

The function of the container of any heat pipe is to isolate the working fluid from the outside environment. It must be leak-proof, maintain the pressure differential across the wall and enable heat to transfer from and into the wick structure. In this particular application one needs the heat pipe to be very flexible under repeated action, easy to bend, resistant to compression force and capable of absorbing thermal stress and vibration. Corrugated containers would be the only ideal

configuration due to the last constraint. Compatability of the container, working fluid and the wick structure is also of significant importance. Due to the above constraints, as well as the selection of Freon R12 as the working fluid, stainless steel is selected for the container. Fortunately, PSM was able to find a corrugated 321 stainless steel tube, which is extremely flexible and is compressible by at least 20% and extendable by 50% of its nominally produced flexible length. Its flexibility is identical to rubber and plastic tubing. Its vacuum rating is considered as ultrahigh vacuum.

IV. METHOD OF FLEXIBLE HEAT PIPE TESTS

A. Introduction

The purpose of this chapter is to report the method and the subsequent results in testing two heat pipes. These two heat pipes were corrugated and flexible at low temperature. The smaller of the two heat pipes (hereafter referred to as heat pipe 1) was constructed of corrugated 321 stainless steel. The larger (hereafter referred to as heat pipe 2) was also of stainless steel of a different flexibility. Both used Freon R12 as the working fluid. Heat pipe 1 weighed 43 grams and contained 10 grams of Freon R12. Both incorporated carbon fiber wicks. The outer diameter of heat pipe 1 was 3/8 inch. Its total lenth, includig end caps, was 18 inches. The outer diameter of heat pipe 2 was 5/8 inch and the length, including end caps, was 22.75 inches. Data on temperature distribution as well as heat transfer capacities for the two pipes were computed for both pipes in three configurations: horizontally, condenser up, condenser down. The application envisioned was with cold weather mittens to transfer heat from the forearm area to finger area of mittens.

B. Experimental Procedure

A schematic diagram of the experimental setup is shown in Fig. 9. A 1/2" plywood box was constructed to protect the pipes during testing and to allow for ease in changing their orientation. The pipes were mounted in the box using standard hardware as shown in Fig. 10. A 6.5" condenser section was constructed of standard 2" PVC with appropriate 40 endcaps of the same material. Holes were drilled in this condenser section to accommodate 1/4" plastic pipe used for outlet and inlet of water to

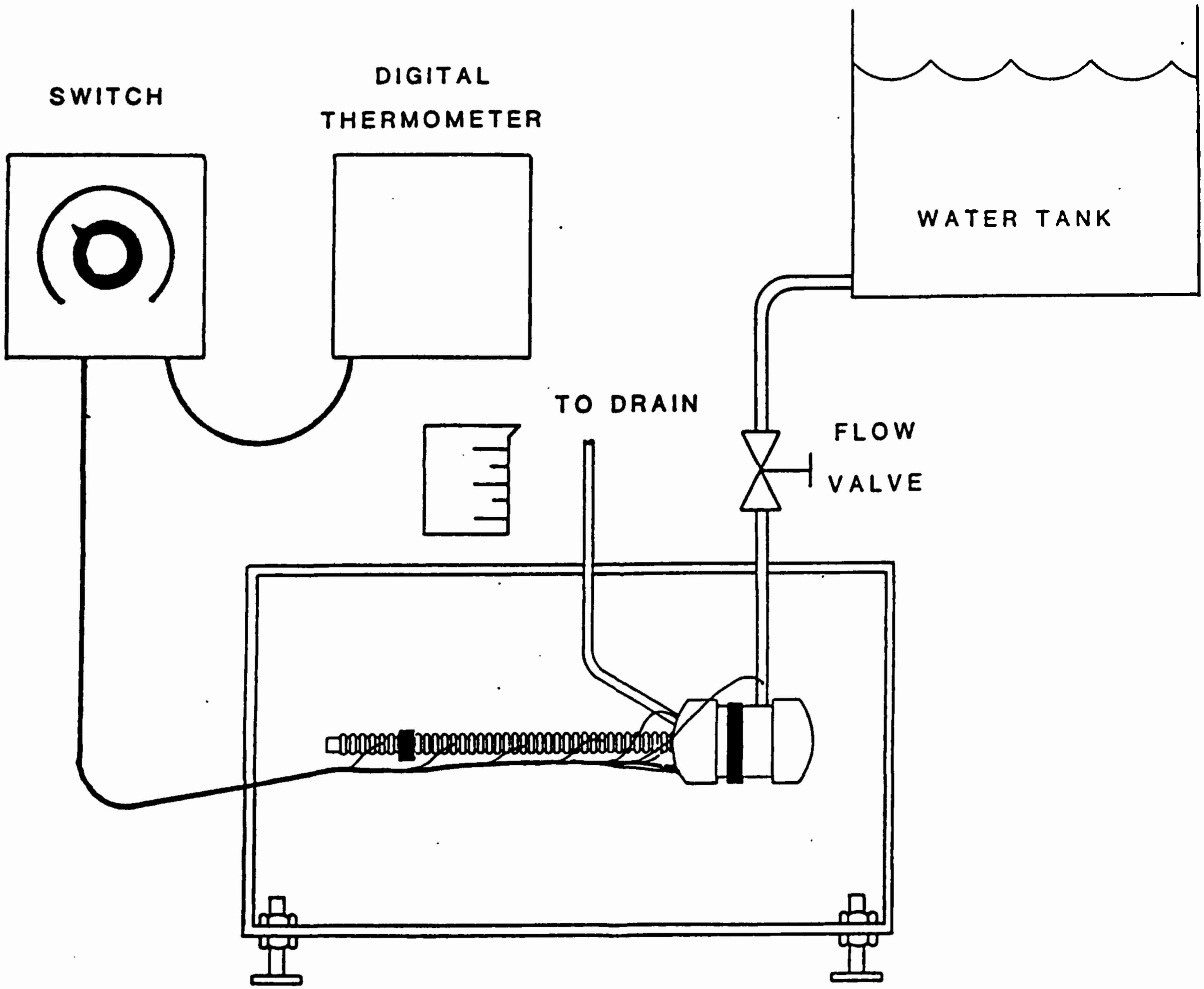
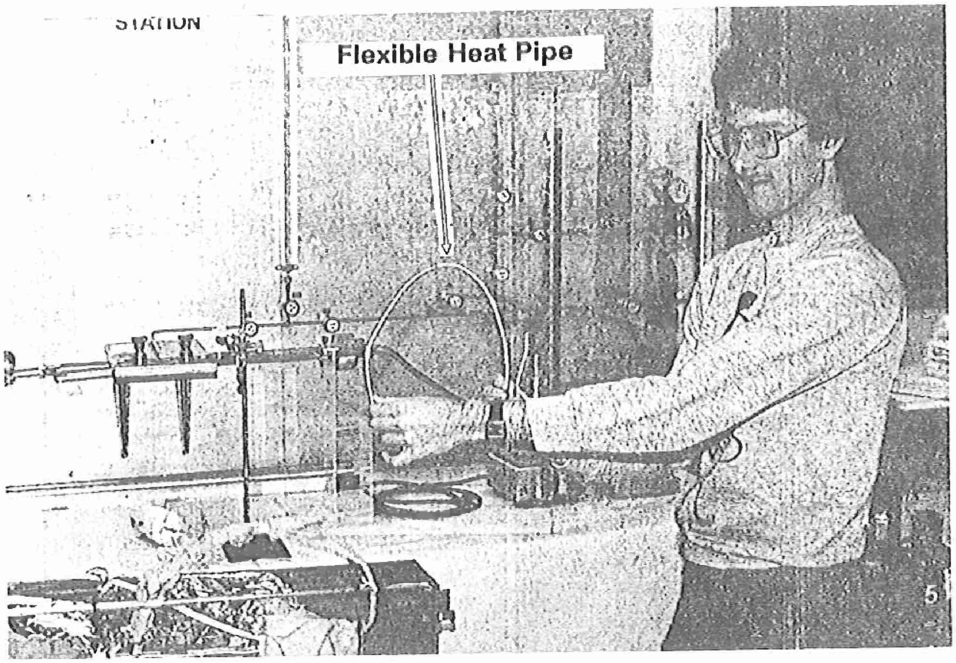


Figure 9. Schematic of Test Set-up



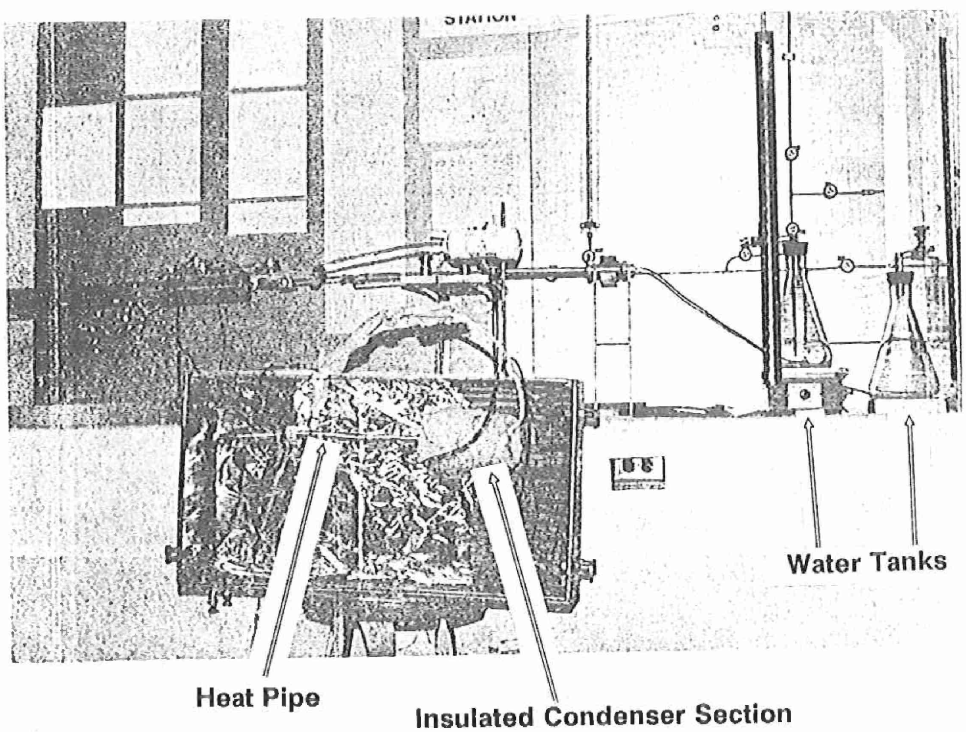


Figure 10. Photographs of Experimental Set-Up and Flexible Heat Pipe

condenser section. In addition, a small hole was required to accommodate theromocouple wires. During the test, the condenser was wrapped with rolled fiberglass insulation to prevent unnecessary error caused by heat exchange with the atmosphere.

To measure the axial temperature distribution of the pipes, thermocouples were mounted along both pipes as shown in Figs. 11-12. These thermocouples were Type T (Copper-Constantan) and were attached to terminals of an "Omega 36" position switch and a "Fluke 2170" digital thermometer. A heating pad was used on the evaporator section of each pipe. In the case of the smaller heat pipe (1) this pad covered all of the evaporator section. In the case of the larger (2), a small section (about 4") of pipe was left exposed to the atmosphere.

To measure heat transport capability of the two pipes, thermocouples were mounted in the inlet tube of each condenser (about 1" above condenser), and at the outlet hole. The inlet pipe was connected to an elevated tank of water. The capacity of the tank was large compared to the volume of water used in a test so effects of changes in static pressure over time were minimal. Water temperatures used ranged from 39°-63°F (4-17°C). A quarter turn valve on the inlet line was used to control flow rate through condenser. By measuring the temperature change between the two thermocouples as well as the mass flow through the condenser, the energy transferred was computed.

Five different cases were used for each pipe. These are as follows:

HEAT PIPE I

Case I - Horizontal, flow rate = 26 mL/min, Tank water 63°F (17°C)

Case II - Horizontal, flow rate = 3.3 mL/min, Tank water 63°F (17°C)

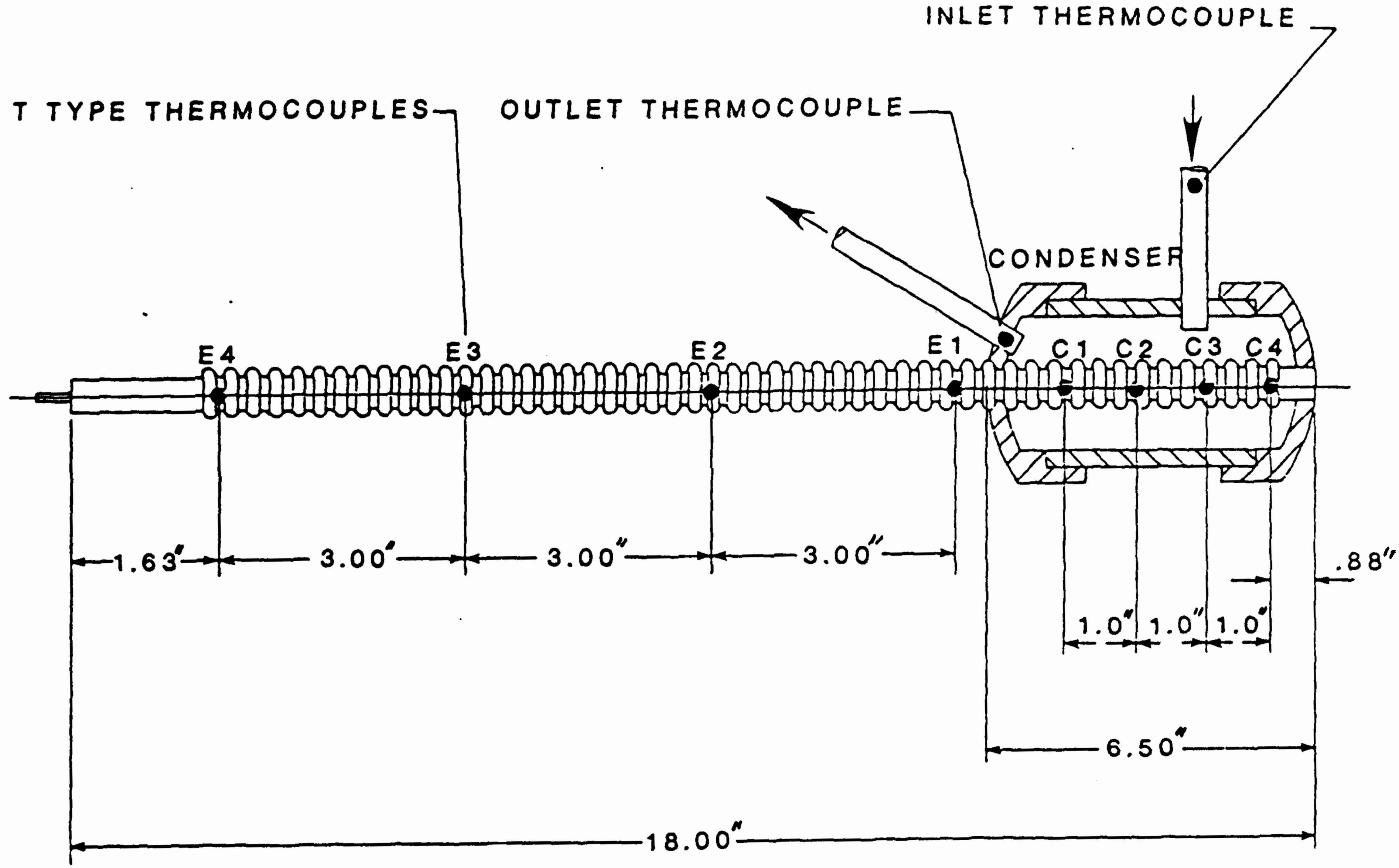


Figure 11. Type 1 - Flexible Heat Pipe

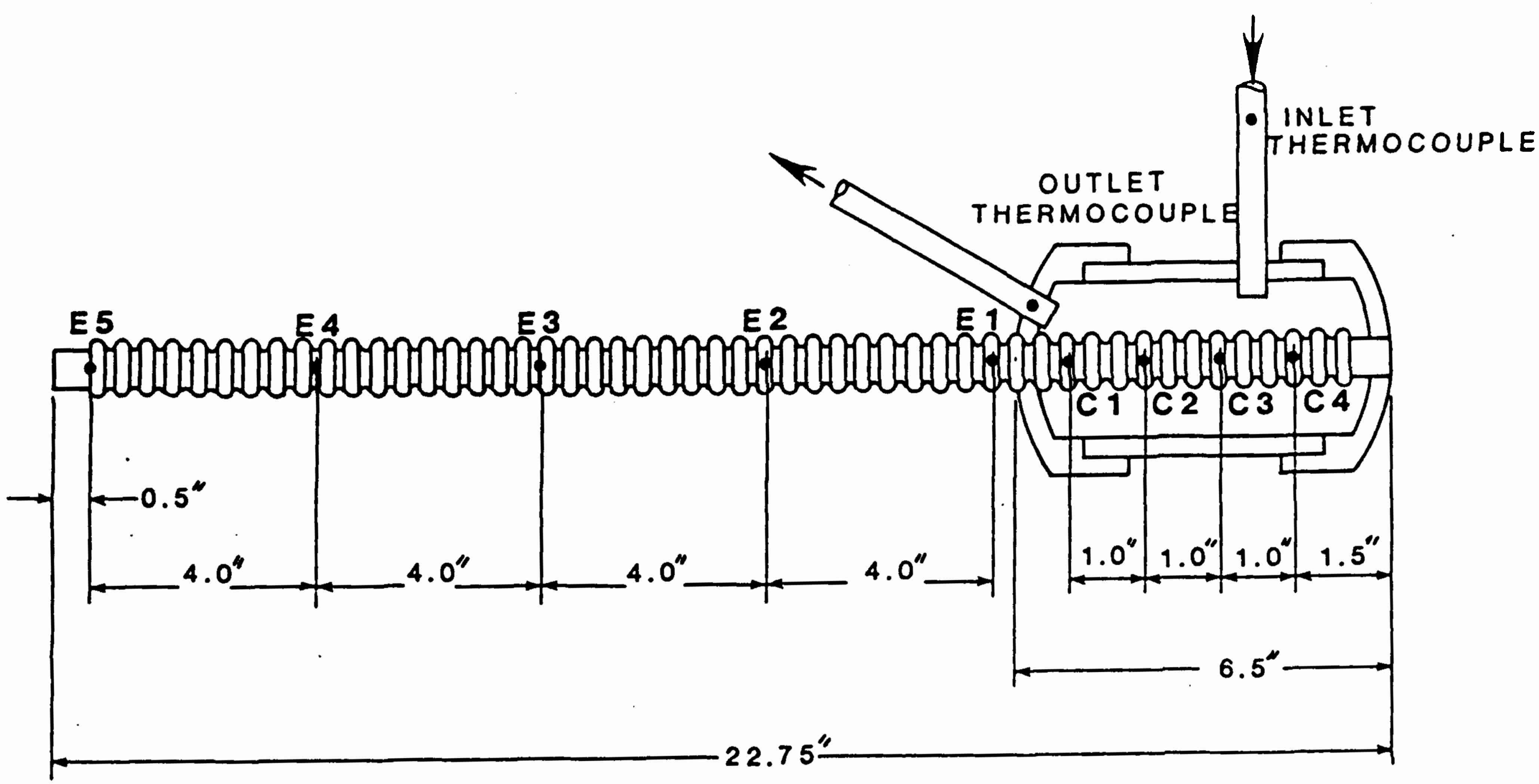


Figure 12. Type 2 - Flexible Heat Pipe

Case III - Vertical, Condenser up, flow rate = 27 mL/min, Tank water 50°F (10°C)

Case IV - Vertical, Condenser up, flow rate = 8.3 mL/min, Tank water 50°F (10°C)

Case V - Vertical, Condenser down, flow rate = 27 mL/min, Tank water 50°F (10°C)

HEAT PIPE 2

Case I - Horizontal, flow rate = 24 mL/min, Tank water 63°F (17°C)

Case II - Horizontal, flow rate = 3.5-6.5 mL/min, Tank water 63°F (17°C)

Case III - Vertical, Condenser up, flow rate = 29 mL/min, Tank water 45°F (7°C)

Case IV - Vertical, Condenser up, flow rate = 2.4 mL/min, Tank water 45°F (7°C)

Case V - Vertical, Condenser down, flow rate = 25 mL/min, Tank water 41°F (5°C)

V. RESULTS

The axial temperature distributions for heat pipe 1 under five different configurations are shown in Tables 1 to 5. Accompanying Figs. 13-17 are plots of temperature vs. length as measured by the thermocouples on the pipe. Temperature differences within the condenser had no physical significance and so are not represented on the plots. Additionally, the heating blanket used caused thermocouples in direct contact with heating blanket elements to show inordinately high values. These, although included in the tables, have been omitted from the graphs.

As shown in Figs. 13-17, heat pipe 1 exhibited nearly isothermal behavior throughout the test in all configurations with the exception of condenser down. Even in this configuration it was still capable of some heat transfer. As shown in Fig. 18, a steady state heat transfer of as much as 4 watts was measured in the horizontal case and positive heat flow in all cases.

The axial temperature distributions for heat pipe 2 under five different configurations are shown in Tables 6 to 10. Accompanying Figs. 19-23 are plots of temperature versus length as measured by the thermocouples on the pipe. Similar data points, as with heat pipe 1, have been ignored in the plots but are represented in the data tables.

As shown in Figs. 19-23 heat pipe 2 exhibited similar isothermal behavior as did pipe 1 under similar configurations. As much as 4 watts were transferred, as shown in Fig. 24. All configurations attempted resulted in positive heat transfer.

TABLE 1. Axial Temperature Distribution and Heat Flow, Heat Pipe 1, Case I.

Case I - the small heat pipe was in a horizontal position with a heating pad around the evaporator section, the flow rate was 26 mL/minute, readings were taken 20 minutes apart. Tap water was used in the condenser.

	Temperature (Celsius)										
Thermo-couple	1	2	3	4	5	6	7	8	9	10	11
Inlet	21.2	18.8	19.0	19.0	19.0	19.2	19.2	19.2	19.2	19.4	19.4
Outlet	24.6	21.4	20.8	21.0	21.2	21.6	21.6	21.8	21.8	22.0	22.0
C1	23.2	20.4	20.0	20.0	20.2	20.2	20.4	20.4	20.4	20.6	20.8
C2	24.2	21.4	20.8	21.0	21.2	21.2	21.4	21.6	21.6	21.6	21.8
C3	23.0	20.2	19.8	20.0	20.2	20.2	20.4	20.4	20.4	20.4	20.6
C4	23.4	20.8	20.2	20.4	20.4	20.6	20.8	20.8	20.8	20.8	21.0
E1	25.0	21.8	21.8	22.2	23.0	23.2	23.2	22.8	22.8	23.8	24.2
E2	27.8	25.2	24.4	25.0	25.8	25.6	25.6	25.2	25.0	26.2	27.0
E3	29.0	26.0	27.8	28.6	28.2	27.8	27.8	26.6	26.6	27.8	28.2
E4	27.4	40.8	43.2	45.6	48.6	50.2	49.6	47.4	46.6	50.6	52.4
Watts	6.14	4.70	3.25	3.60	3.97	4.34	4.34	4.69	4.70	4.70	4.70

Note: See figures 11 and 12 for placement of thermocouples for all tables.

TABLE 2. Axial Temperature Distribution and Heat Flow, Heat Pipe 1, Case II

Case II - the small heat pipe was in a horizontal position with a heating pad around the evaporator section, the flow rate was 3.3 mL/minute, readings were taken 20 minutes apart. Tap water was used in the condenser.

	Temperature (Celsius)										
Thermo-couple	1	2	3	4	5	6	7	8	9	10	11
Inlet	21.0	21.4	21.6	21.6	21.6	21.8	21.6	21.8	21.8	22.0	21.6
Outlet	25.2	26.8	27.6	28.0	28.2	28.4	28.4	28.4	28.2	28.0	28.2
C1	22.2	23.8	24.8	25.2	25.6	25.8	25.8	25.8	25.6	25.8	26.6
C2	25.2	27.2	27.6	28.0	28.2	28.4	28.4	28.2	28.0	27.6	28.0
C3	22.2	23.8	24.6	25.0	25.2	25.6	25.6	25.6	25.6	27.2	27.6
C4	23.8	25.6	26.2	26.8	27.0	27.2	27.2	27.2	27.0	27.6	27.8
E1	26.6	28.6	28.8	28.6	28.6	29.6	29.0	28.8	28.6	29.0	29.0
E2	29.0	31.4	31.0	30.4	30.4	31.8	30.8	30.4	30.2	30.8	30.6
E3	29.8	32.4	31.4	31.2	30.8	32.0	31.4	31.4	30.8	31.2	30.8
E4	47.8	51.4	44.2	41.6	40.2	43.2	40.0	39.2	37.8	40.6	41.8
Watts	0.95	1.22	1.36	1.45	1.49	1.49	1.54	1.49	1.45	1.36	1.49

TABLE 3. Axial Temperature Distribution and Heat Flow, Heat Pipe 1, Case III

Case III - the small heat pipe was in a vertical position with the evaporator below the condenser, the evaporator was covered with a heating pad, the flow rate was 27 mL/minute. Ice water was used in the condenser.

Temperature (Celsius)

Thermo-couple	1	2	3	4	5	6	7	8	9	10
Inlet	14.8	12.3	11.8	11.8	11.6	11.6	11.6	11.6	11.5	11.6
Outlet	17.2	14.4	13.6	13.4	13.2	13.2	13.2	13.2	13.2	13.4
C1	24.2	20.4	18.6	17.8	17.4	17.4	17.2	17.2	16.8	17.2
C2	24.6	21.8	20.4	19.8	19.4	19.6	19.6	19.6	19.6	20.2
C3	28.8	25.6	23.4	22.6	22.1	22.0	21.6	21.4	21.2	21.2
C4	28.6	25.6	23.6	22.8	22.4	22.8	22.8	22.6	22.6	23.0
E1	26.0	24.6	23.2	23.0	23.2	23.2	23.2	24.2	23.2	24.6
E2	28.2	26.2	25.2	24.8	25.2	25.2	26.2	27.4	26.2	27.8
E3	28.4	26.4	25.4	25.2	25.6	25.8	26.6	27.6	26.4	28.2
E4	27.8	25.6	24.6	24.2	24.4	24.4	24.8	25.2	24.6	25.8
Watts	4.50	3.94	3.38	3.00	3.00	3.00	3.00	3.00	3.19	3.19

TABLE 4. Axial Temperature Distribution and Heat Flow, Heat Pipe 1, Case IV

Case IV - the small heat pipe was in a vertical position with the evaporator below the condenser, the evaporator was covered by a heating pad, the flow rate was 8.3 mL/minute, readings were taken 20 minutes apart. Ice water was used in the condenser.

	Temperature (Celsius)								
Thermo-couple	1	2	3	4	5	6	7	8	9
Inlet	14.8	16.2	17.2	17.8	18.2	18.2	18.2	18.6	19.0
Outlet	20.0	20.2	22.2	22.2	23.2	22.8	23.2	23.8	24.8
C1	20.6	21.6	22.6	23.0	23.4	23.8	24.2	24.6	24.6
C2	24.2	25.2	25.8	26.8	26.8	27.8	28.0	28.2	28.0
C3	22.2	23.0	23.6	24.2	24.6	24.8	24.8	25.4	25.4
C4	27.8	28.8	29.2	30.4	29.8	31.6	31.6	31.8	31.2
E1	29.2	29.8	30.2	31.4	31.6	33.2	33.0	33.4	33.2
E2	33.0	33.4	33.6	34.8	34.6	36.6	36.0	36.8	36.6
E3	32.6	32.0	33.4	34.6	34.2	36.2	36.0	36.4	35.8
E4	31.8	32.3	32.6	34.0	33.2	35.4	35.2	35.4	35.0
Watts	1.92	1.47	1.84	1.62	1.84	1.69	1.84	1.92	2.14

TABLE 5. Axial Temperature Distribution and Heat Flow, Heat pipe 1, Case V

Case V - the small heat pipe was in a vertical position with the condenser below the evaporator, the evaporator was covered by a heating pad, the flow rate was 27 mL/minute, readings were taken 20 minutes apart. Ice water was used in condenser.

	Temperature (Celsius)									
Thermo-couple	1	2	3	4	5	6	7	8	9	10
Inlet	12.2	12.0	12.2	12.2	12.4	12.4	12.6	12.8	12.8	13.0
Outlet	15.2	13.0	12.8	12.8	12.8	13.2	13.2	13.4	13.2	13.4
C1	13.6	12.2	12.2	12.2	12.4	12.6	12.6	12.8	12.8	13.0
C2	13.6	12.2	12.2	12.2	12.4	12.4	12.6	12.8	12.8	13.0
C3	13.2	12.2	12.4	12.4	12.6	12.8	12.8	13.0	13.0	13.0
C4	12.6	12.0	12.0	12.2	12.2	12.4	12.6	12.6	12.8	12.8
E1	34.8	36.6	34.6	36.4	35.2	36.4	35.8	36.6	35.6	37.6
E2	65.4	64.0	60.2	64.8	61.8	65.0	62.6	64.8	63.4	66.2
E3	68.6	67.6	63.8	68.2	65.6	68.4	66.0	68.2	65.2	67.2
E4	62.6	59.8	56.8	60.0	57.8	60.0	57.6	59.6	56.8	58.8
Watts	5.63	1.88	1.12	1.13	0.75	1.50	1.12	1.12	0.75	0.75

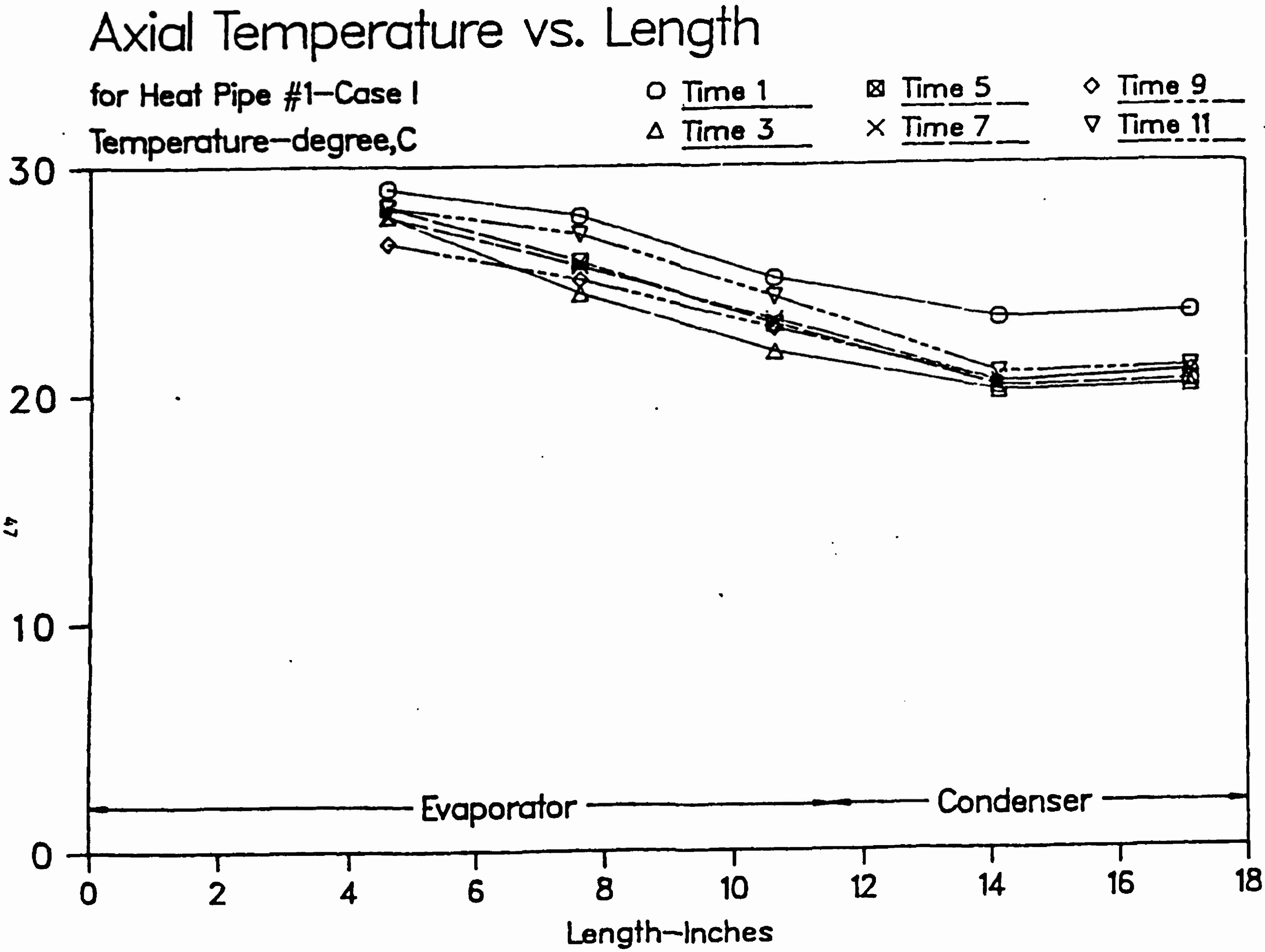


Figure 13. **Axial Temperature vs. Length for Heat Pipe #1 - Case 1**

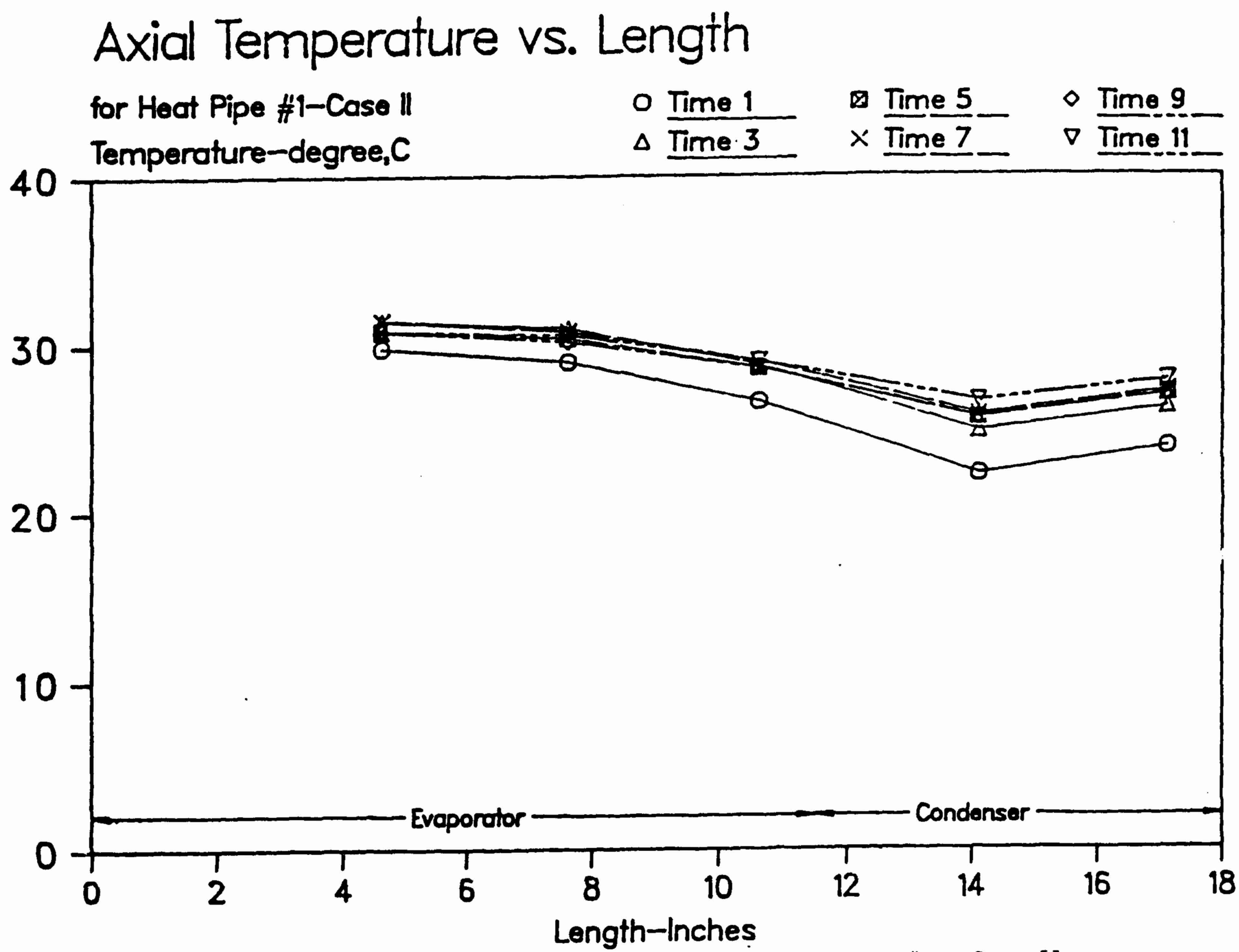


Figure 14. Axial Temperature vs. Length for Heat Pipe #1 - Case II

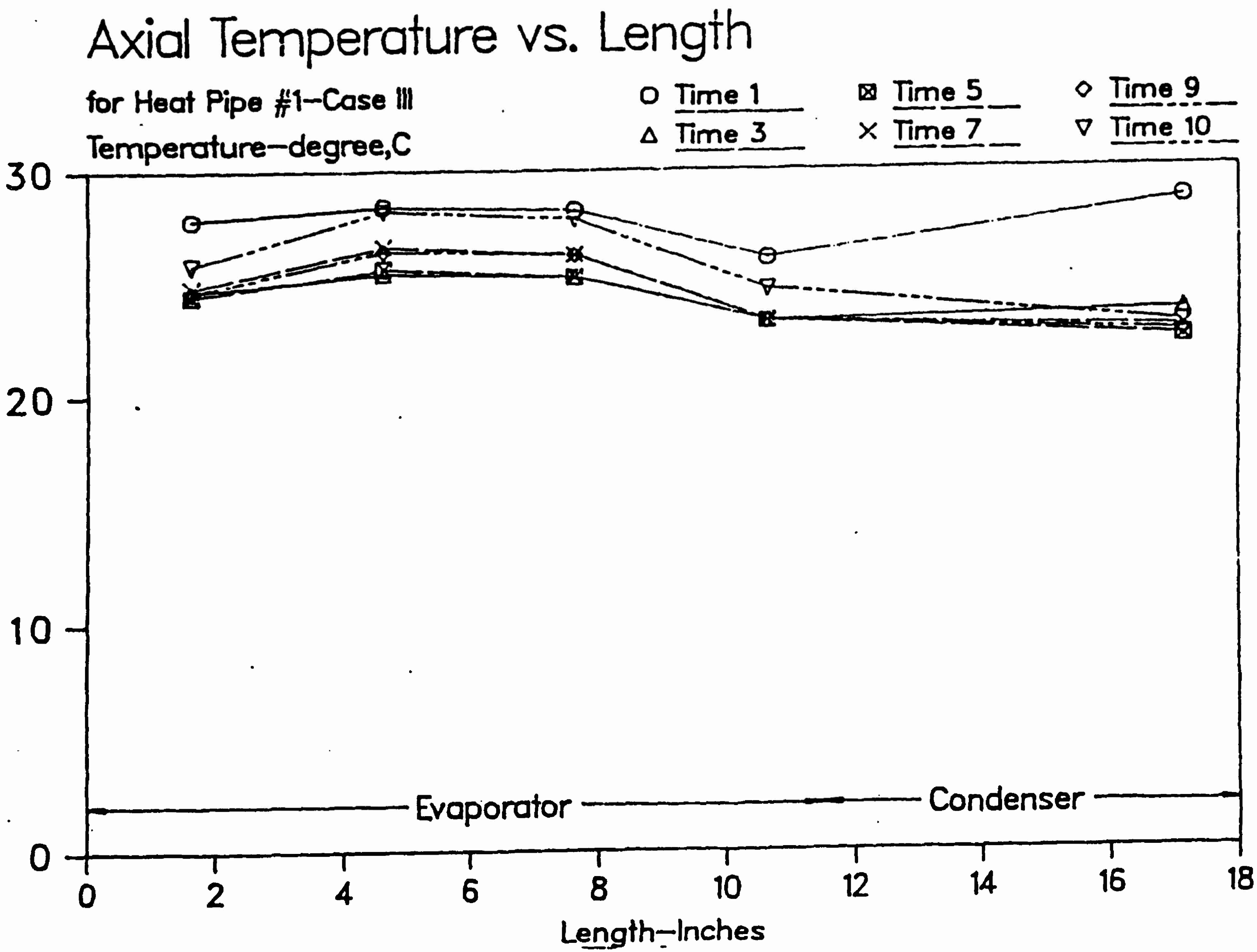


Figure 15. Axial Temperature vs. Length for Heat Pipe #1 - Case III

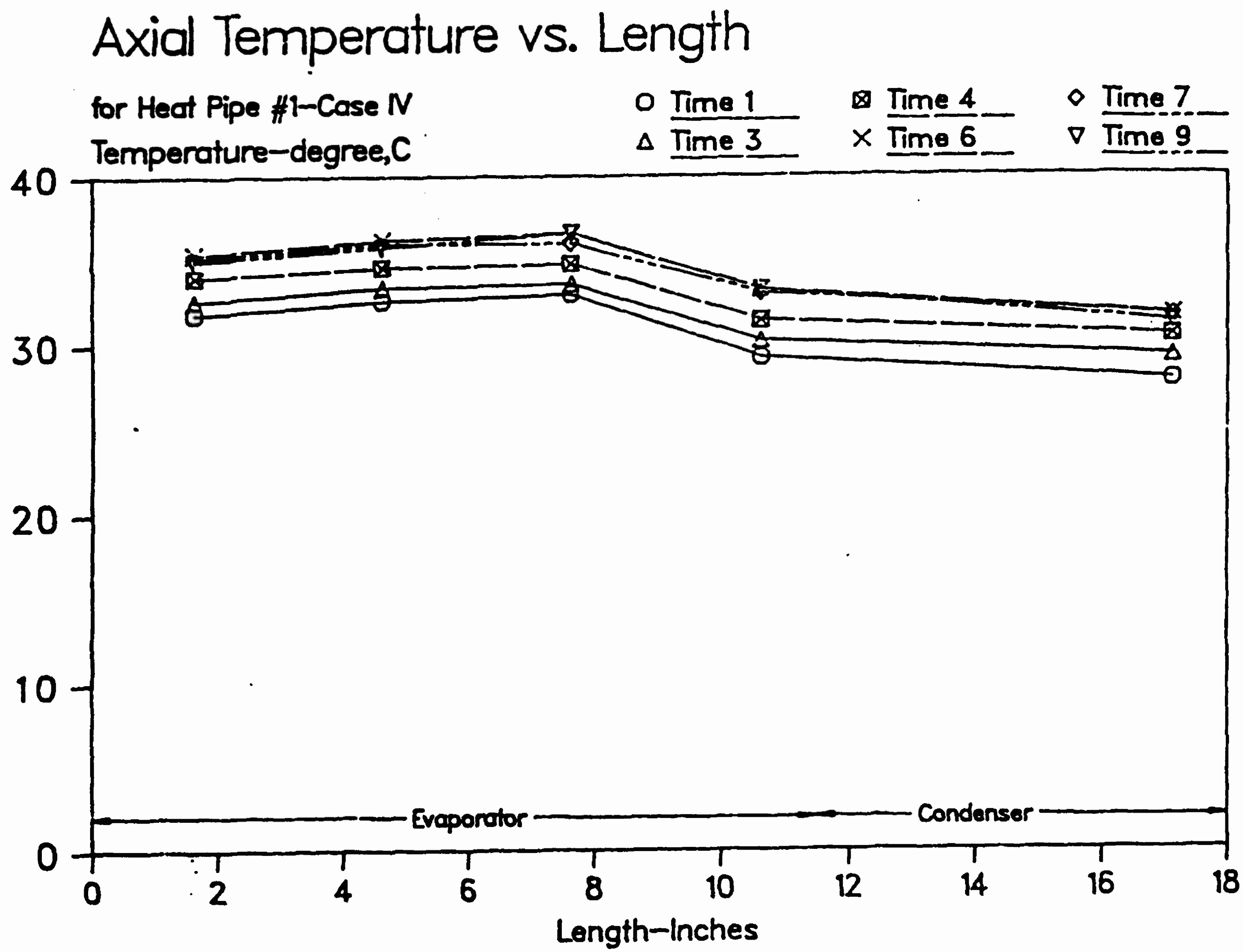


Figure 16. Axial Temperature vs. Length for Heat Pipe #1 - Case IV

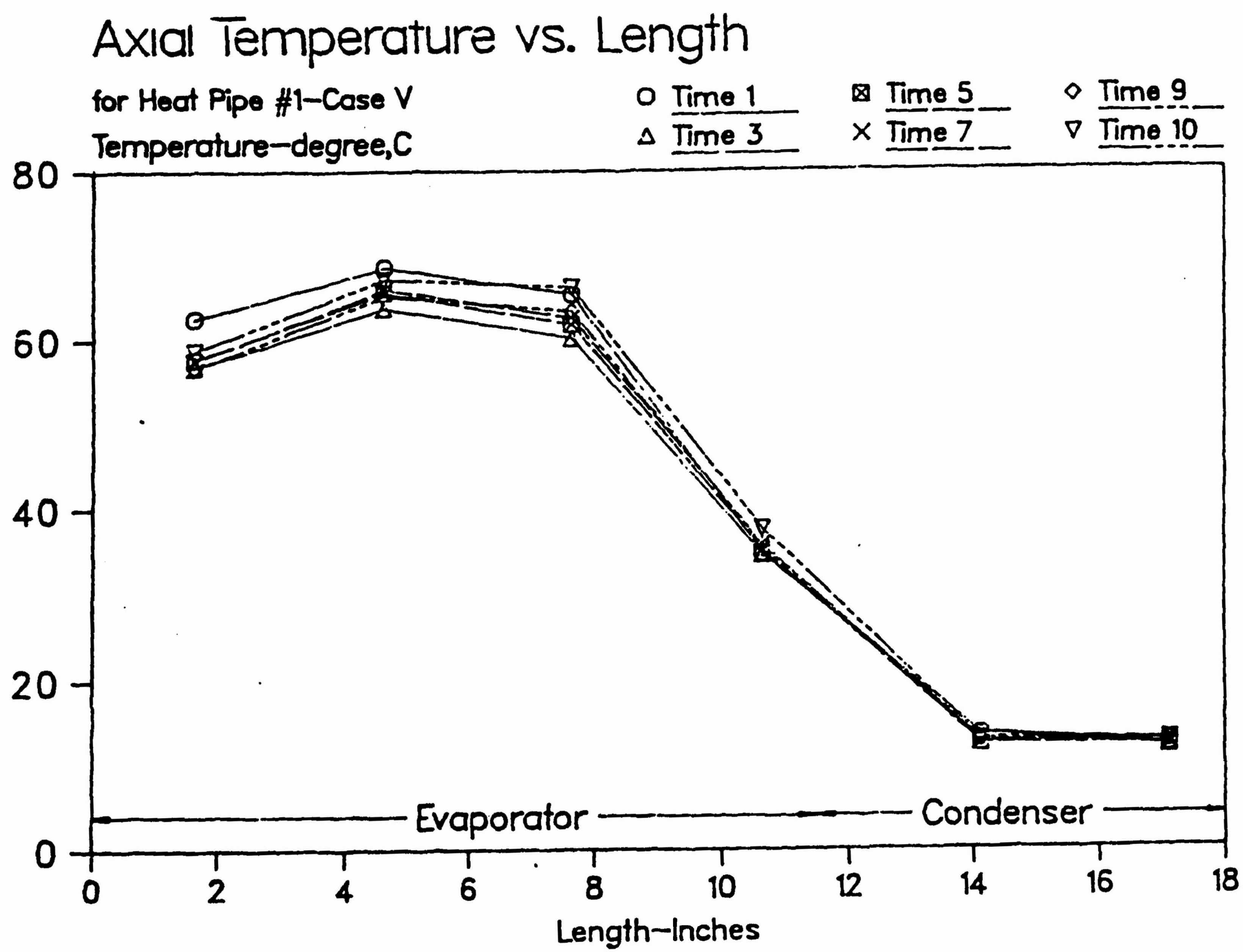


Figure 17. <u>Axial Temperature vs. Length for Heat Pipe #1 - Case V</u>

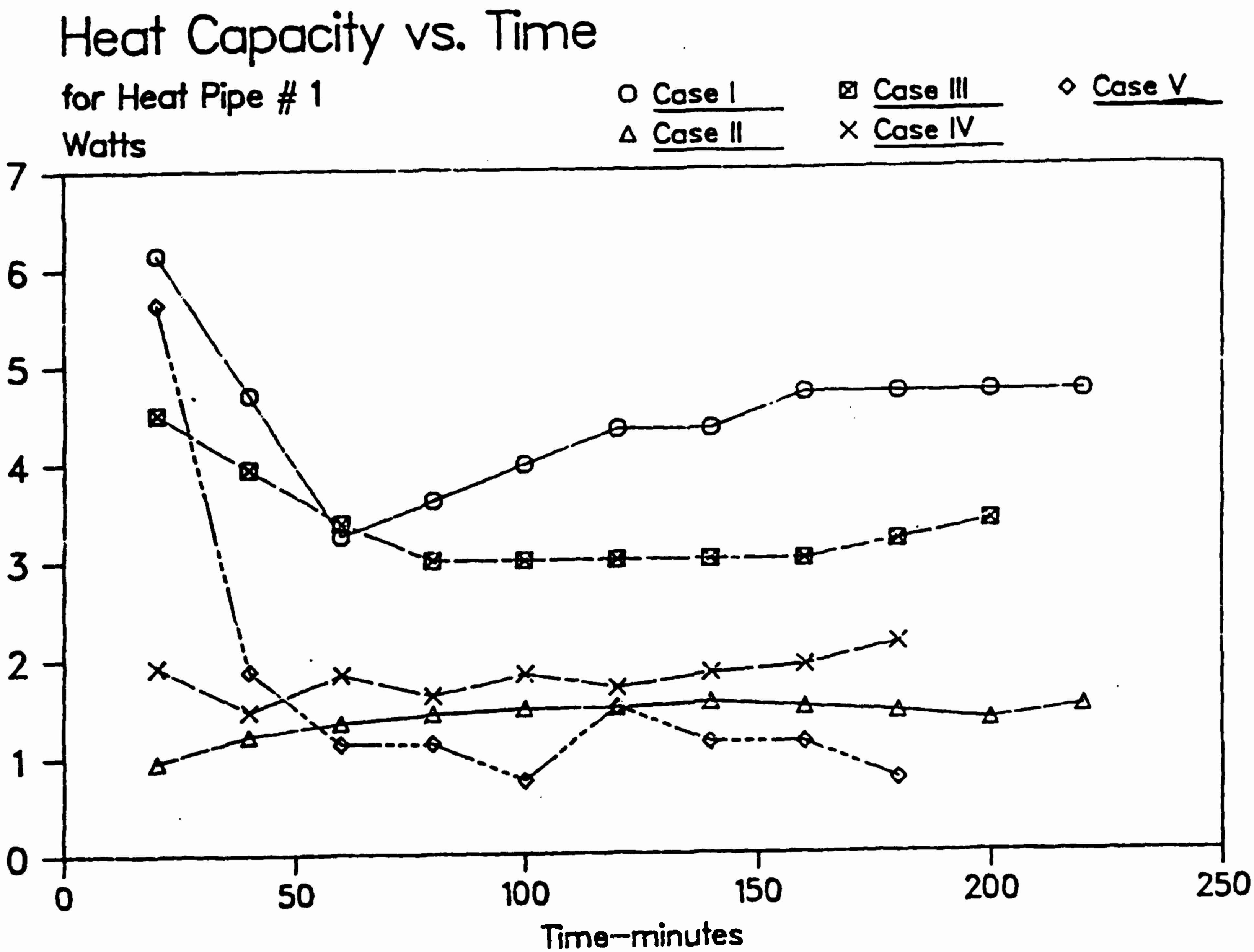


Figure 18. **Heat Capacity vs. Time for Heat Pipe #1**

TABLE 6. Axial Temperature Distribution and Heat Flow, Heat Pipe 2, Case I

Case I - larger of two heat pipes was mounted horizontally, heating pad was placed on evaporator section, the flow rate through condenser was 24 mL/min. Approximately 4" of pipe was exposed to atmosphere between evaporator and condenser, readings were taken 10 minutes apart, tap water was used in the condenser.

	Temperature (Celsius)							
Thermo-couple	1	2	3	4	5	6	7	8
Inlet	18.6	17.6	17.6	17.6	17.8	17.8	17.8	17.8
Outlet	21.2	20.2	20.0	19.4	20.4	19.6	19.6	19.8
C1	20.4	19.6	19.2	19.2	19.2	19.2	19.2	19.2
C2	21.0	20.2	19.8	19.6	19.8	19.6	20.0	20.0
C3	20.8	20.0	19.4	19.6	19.8	19.2	19.8	20.0
C4	21.0	20.4	20.0	19.8	19.8	19.8	20.0	20.0
E1	22.6	22.2	22.0	22.0	21.6	21.6	21.6	21.8
E2	22.8	22.4	22.4	22.2	21.8	21.6	22.0	22.6
E3	25.8	25.4	26.4	26.4	24.8	25.0	25.2	26.2
E4	31.8	31.2	34.4	34.4	30.6	31.0	31.6	33.2
E5	25.8	25.4	28.4	28.0	26.4	26.0	26.2	28.2
Watts	4.34	4.34	4.00	3.00	4.34	3.00	3.00	3.34

TABLE 7. Axial Temperature Distribution and Heat Flow, Heat Pipe 2, Case II

Case II - heat pipe was in horizontal position, heating pad was placed on evaporator section, flow rate through condenser was 3.5 mL/min for first 4 readings and 6.5 mL/min for last 2. Tap water in condenser. A 4" section of pipe next to condenser end was exposed, 10 minute intervals between readings.

	Temperature (Celsius)						
Thermo-couple	1	2	3	4	5	6	7
Inlet	20.0	20.4	20.8	21.0	21.2	20.6	20.8
Outlet	23.0	24.0	21.8	21.4	23.8	25.4	25.8
C1	21.0	21.8	22.8	23.6	24.2	23.8	24.0
C2	22.4	23.4	24.6	25.2	25.8	24.8	25.4
C3	22.2	23.2	24.4	25.0	25.8	24.8	25.4
C4	22.2	23.2	24.0	24.8	25.4	24.6	25.2
E1	24.2	25.0	25.6	26.2	27.0	26.2	26.6
E2	24.6	25.6	26.0	26.6	27.2	26.6	26.8
E3	28.4	29.2	29.2	30.0	31.4	30.8	30.0
E4	36.4	36.4	35.6	36.2	39.2	38.8	36.2
E5	29.2	29.8	29.4	34.4	36.6	34.6	30.8
Watts	0.73	0.88	0.24	0.09	0.63	2.17	2.23

Comment: During test it was noticed by experimenter that flow had stopped. Flow was reset but experimenter had difficulty matching previous flow rate due to extremely small values.

TABLE 8. Axial Temperature Distribution and Heat Flow, Heat Pipe 2, Case III

Case III - heat pipe was in vertical position, condenser on top, flow rate 29 mL/min, ice water in condenser. Heating pad was applied to evaporator section. Same 4" open section as previous used. 10 minute intervals between readings.

	Temperature (Celsius)							
Thermo-couple	1	2	3	4	5	6	7	8
Inlet	12.4	12.6	12.6	12.6	12.6	12.6	12.6	12.6
Outlet	14.4	14.6	14.6	14.6	14.6	14.6	14.8	14.8
C1	14.6	14.8	14.6	14.8	14.8	14.8	14.8	14.8
C2	14.2	14.2	14.2	14.4	14.4	14.4	14.4	14.6
C3	15.8	16.0	15.8	15.8	15.8	15.8	15.8	15.8
C4	16.2	16.4	16.2	16.2	16.0	16.0	16.2	16.2
E1	16.8	16.8	16.6	16.6	16.6	16.4	16.4	16.6
E2	17.2	17.0	16.8	16.6	16.6	16.4	16.4	16.6
E3	21.6	21.2	21.2	21.2	20.0	20.0	20.0	20.4
E4	21.0	20.8	20.8	21.0	20.4	20.2	20.2	20.4
Watts	4.03	4.03	4.03	4.03	4.03	4.03	3.63	3.63

TABLE 9. Axial Temperature Distribution and Heat Flow, Heat Pipe 2, Case IV

Case IV - heat pipe was in vertical position, condenser up, ice water around condenser, flow rate through condenser was 2.4 mL/min. Interval between readings was 10 minutes. Same 4" open section. Heat pad was applied to evaporator section.

	Temperature (Celsius)							
Thermo-couple	1	2	3	4	5	6	7	8
Inlet	16.0	17.4	19.8	20.6	20.0	21.2	21.4	21.8
Outlet	16.8	18.6	20.0	20.8	21.4	22.0	22.6	23.0
C1	16.6	18.2	20.6	21.6	22.2	22.8	23.2	23.6
C2	15.8	17.6	20.2	21.2	21.8	22.4	23.0	23.4
C3	17.0	18.2	20.2	21.0	21.6	22.2	22.6	23.0
C4	17.6	18.8	21.0	22.2	22.6	23.2	23.8	24.0
E1	17.8	19.2	21.2	21.8	22.6	23.2	23.6	23.8
E2	17.8	19.2	21.2	22.0	22.6	23.2	23.6	23.8
E3	21.8	22.8	23.8	24.0	24.2	24.8	26.4	27.6
E4	22.0	23.6	25.4	26.8	26.8	27.2	28.4	28.2
E5	21.2	22.8	24.8	25.6	26.0	25.8	27.0	27.2
Watts	0.13	0.20	0.03	0.03	0.10	0.13	0.20	0.20

TABLE 10. Axial Temperature Distribution and Heat Flow, Heat Pipe 2, Case V

Case V - heat pipe was in vertical position, condenser down, ice water around condenser, heating pad was applied to evaporator section. Same 4" open section as above, flow rate 25 mL/min. Interval between readings was 10 minutes.

	Temperature (Celsius)							
Thermo-couple	1	2	3	4	5	6	7	8
Inlet	12.0	12.2	12.2	12.0	12.2	12.0	12.0	12.2
Outlet	13.0	12.6	12.6	12.2	12.6	12.6	12.4	12.6
C1	12.6	12.6	12.2	12.2	12.4	12.4	12.2	12.2
C2	12.2	12.6	12.4	12.4	12.2	12.2	12.2	12.4
C3	12.6	12.6	12.6	12.6	12.4	12.4	12.6	12.6
C4	12.4	12.2	12.4	12.2	12.2	12.2	12.4	12.2
E1	15.2	15.0	15.4	15.8	15.8	15.8	16.2	16.2
E2	20.0	20.0	20.2	20.6	21.0	21.4	22.0	22.8
E3	33.8	35.0	35.8	37.2	38.0	38.6	38.6	39.6
E4	63.6	64.8	64.6	64.8	65.8	66.4	65.2	67.0
E5	48.8	49.6	49.6	49.6	50.2	50.4	49.8	50.4
Watts	1.74	0.70	0.70	0.35	0.70	1.05	0.70	0.70

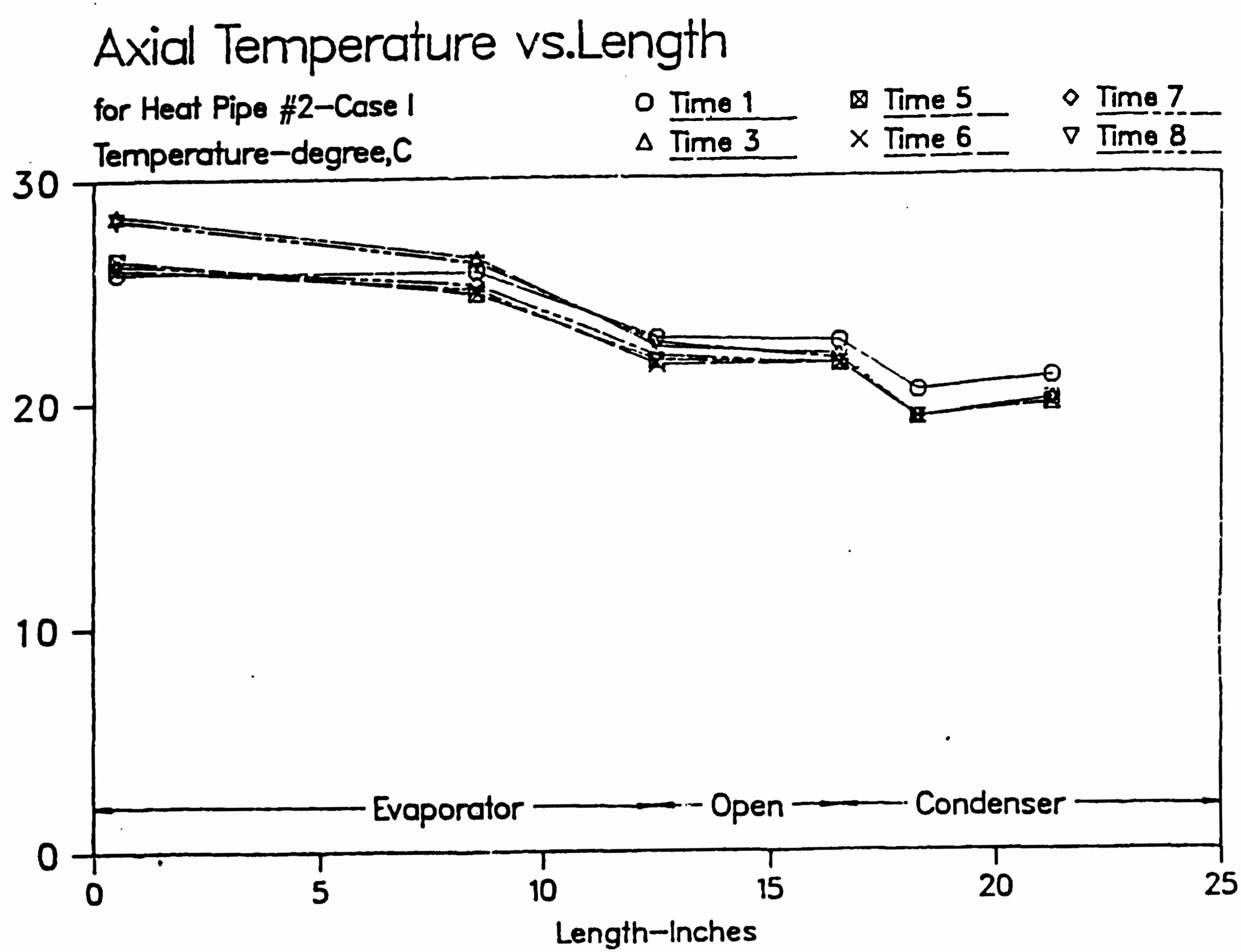


Figure 19. <u>Axial Temperature vs. Length for Heat Pipe #2 - Case I</u>

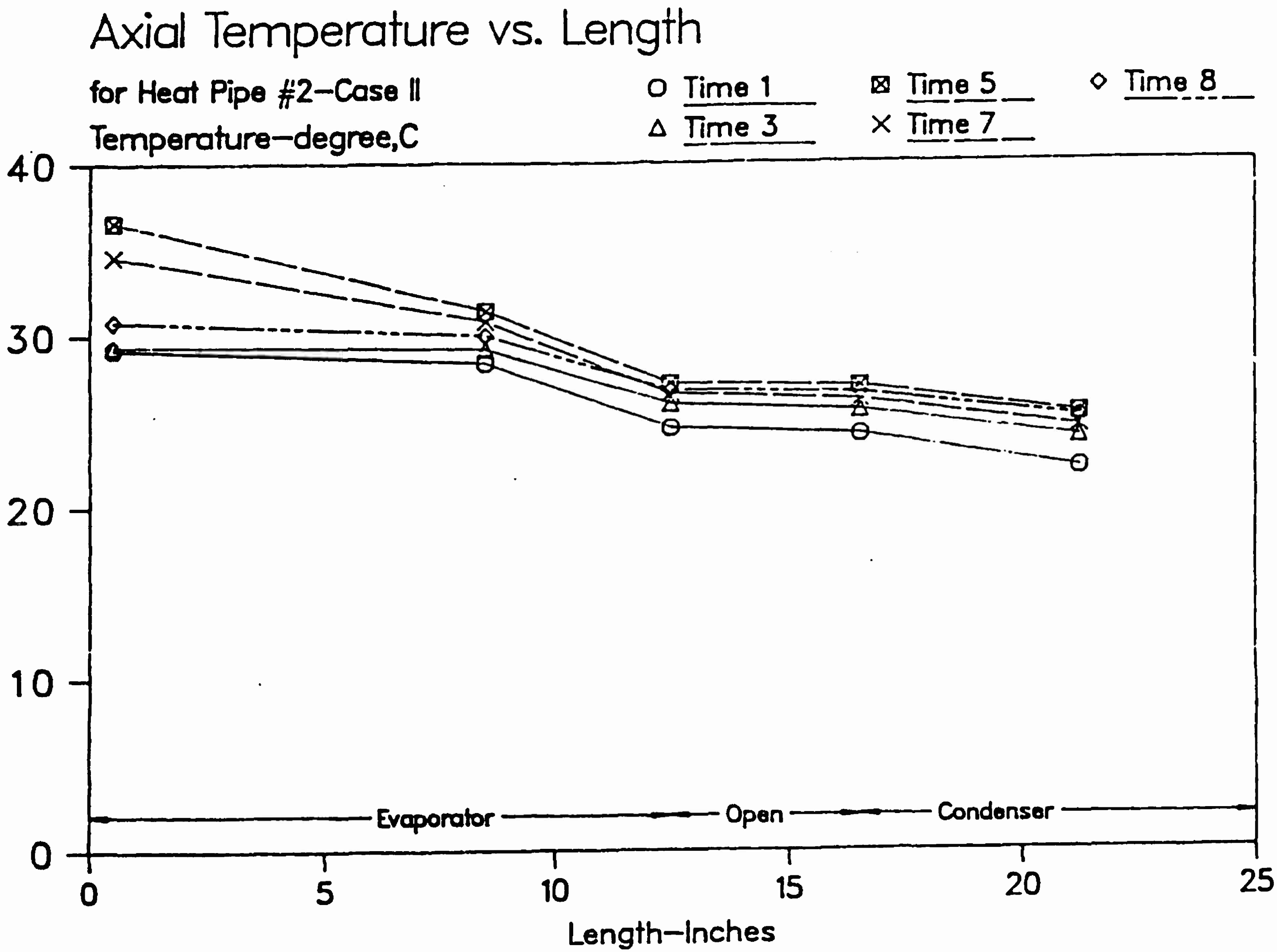


Figure 20. Axial Temperature vs. Length for Heat Pipe #2 - Case II

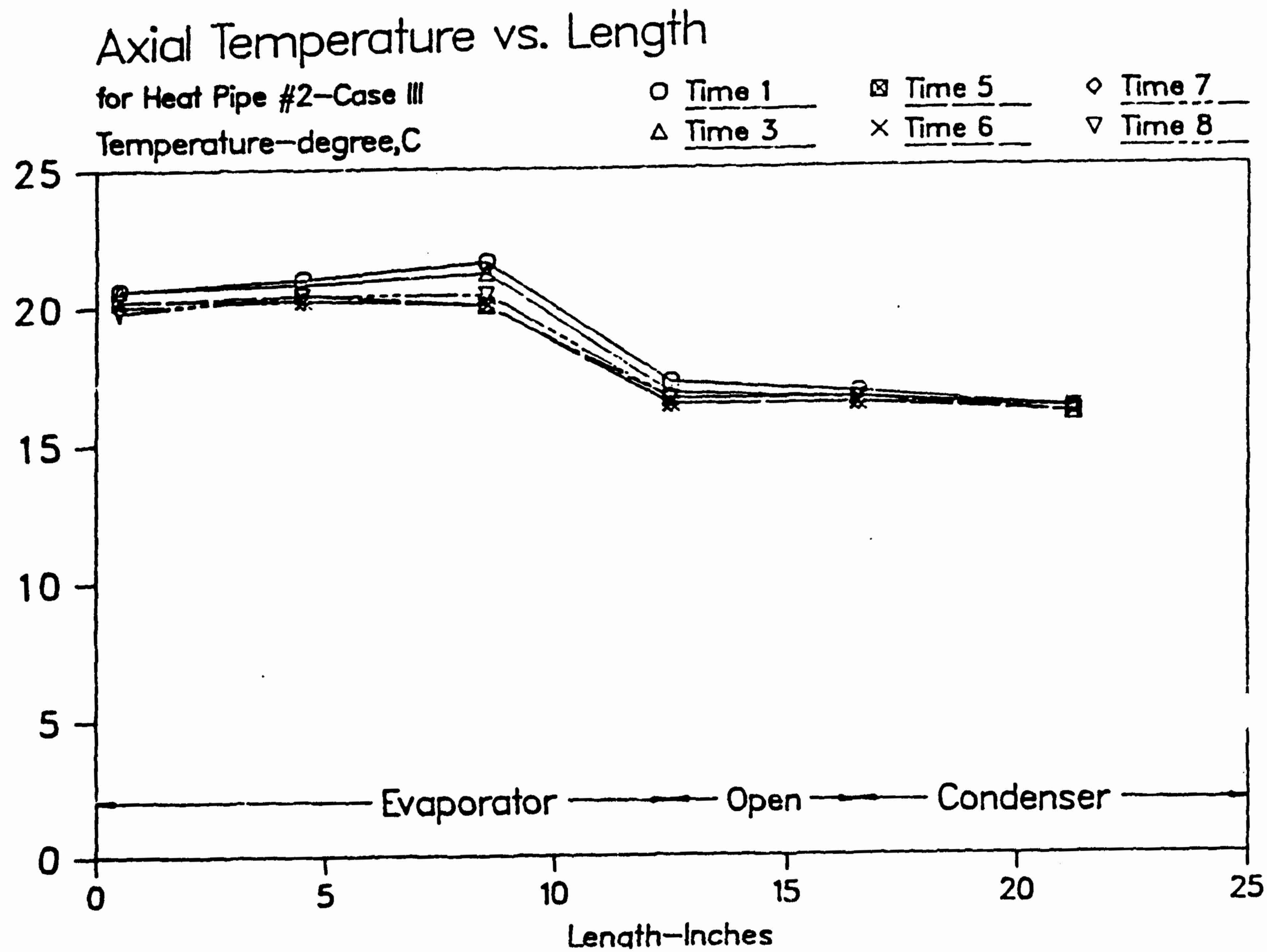


Figure 21. Axial Temperature vs. Length for Heat Pipe #2 - Case III

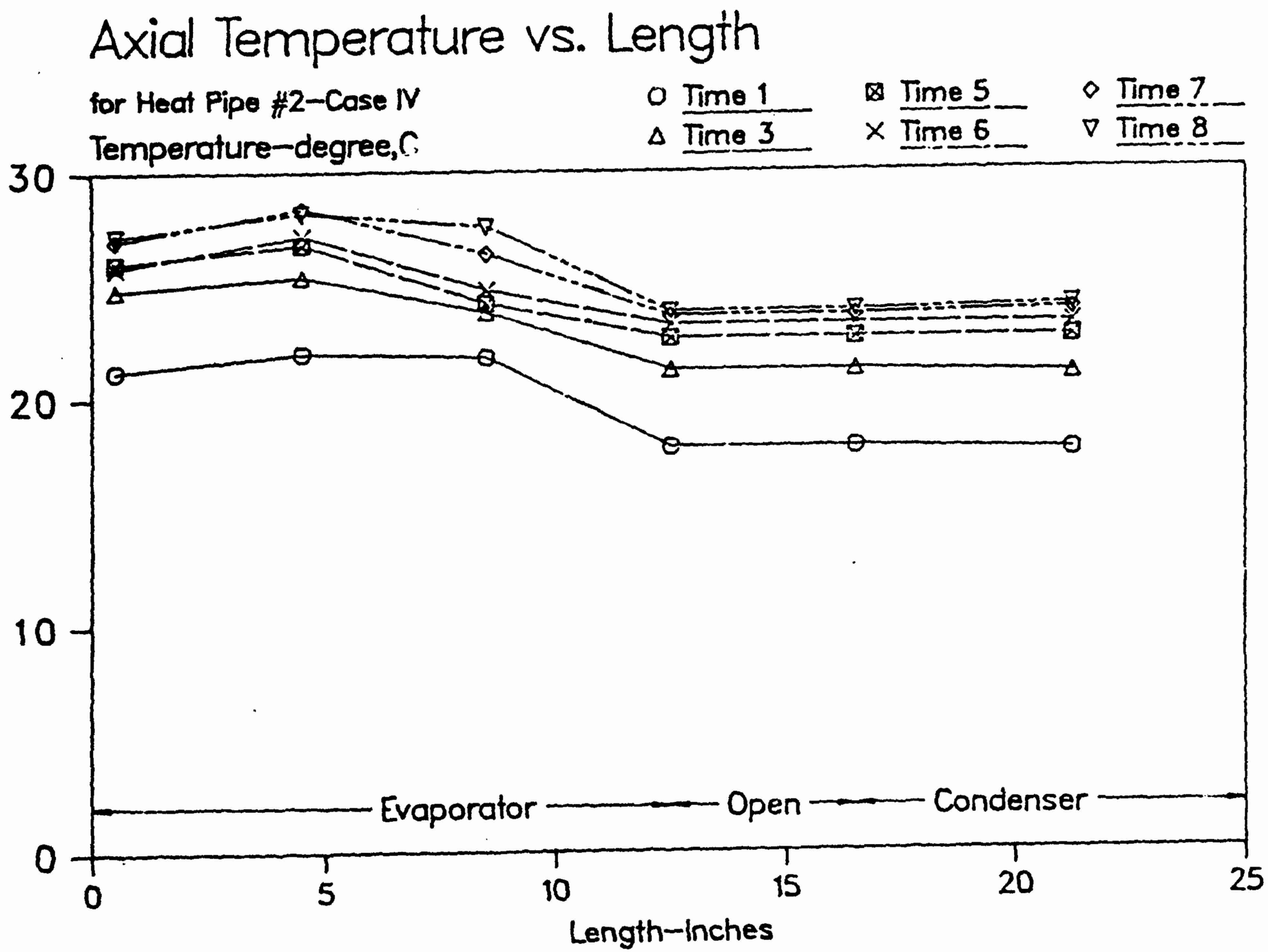


Figure 22. **Axial Temperature vs. Length for Heat Pipe #2 - Case IV**

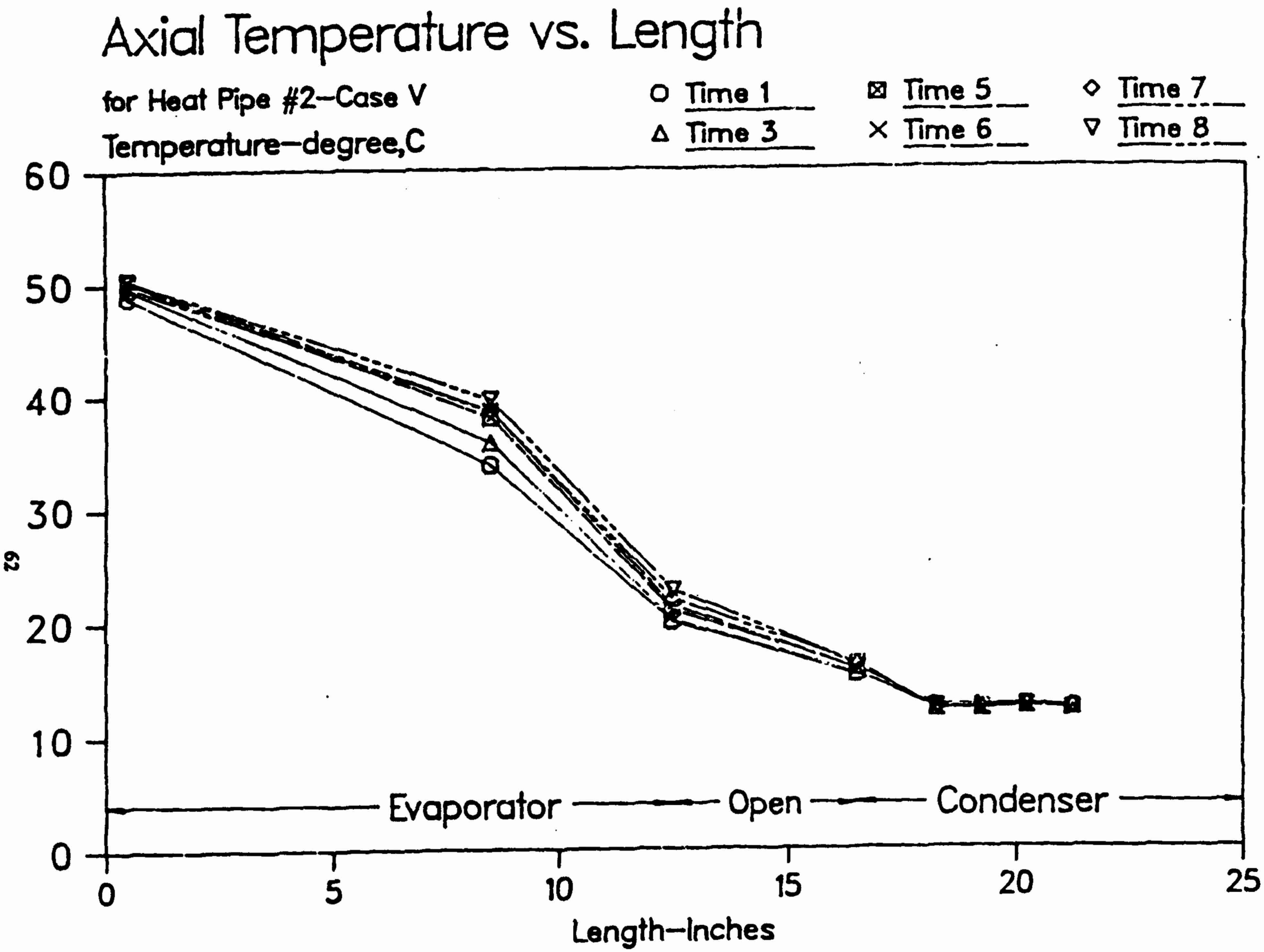


Figure 23. Axial Temperature vs. Length for Heat Pipe #2 - Case V

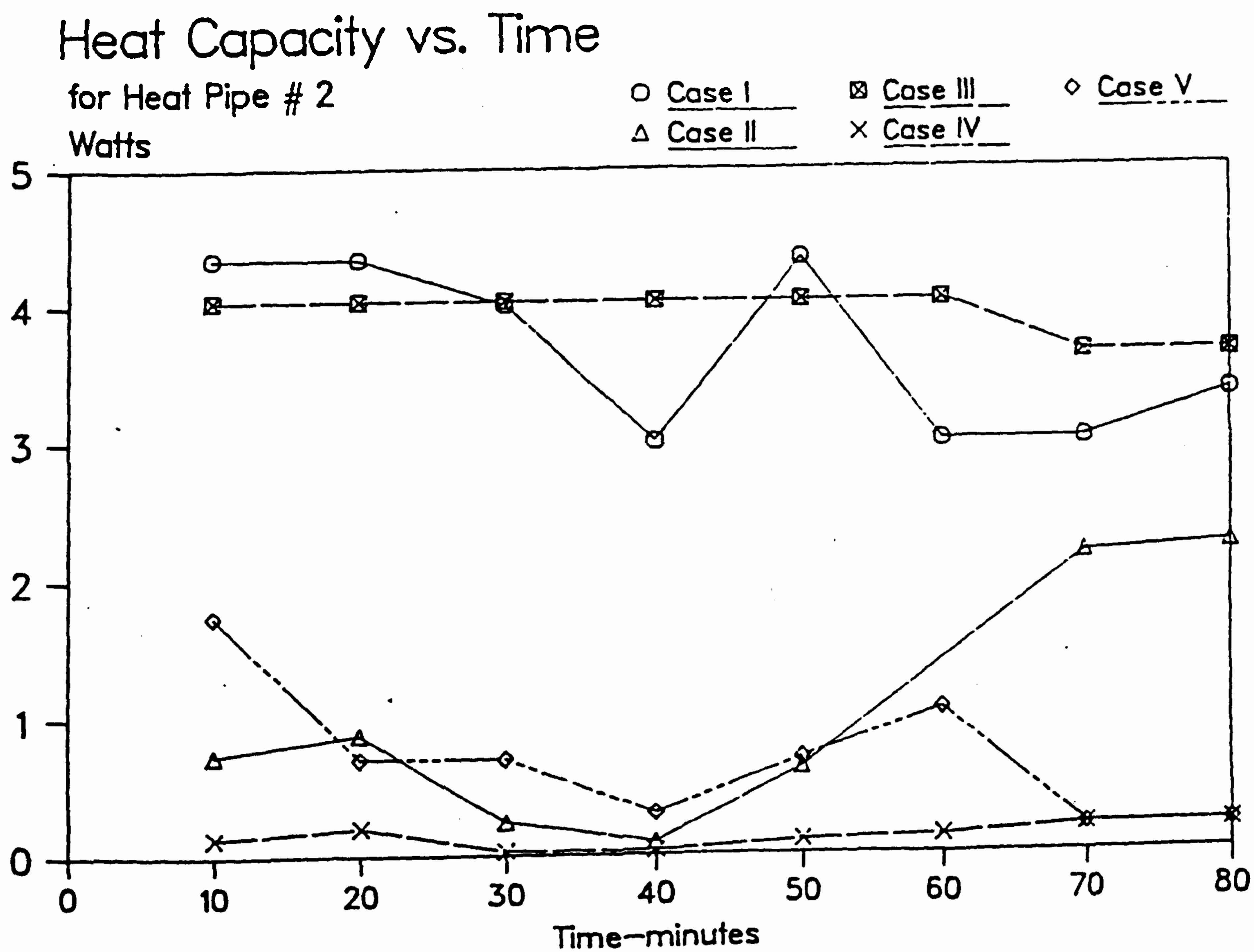


Figure 24. Heat Capacity vs. Time for Heat Pipe #2

VI. CONCLUSIONS

Both heat pipes performed in all three different attitudes: horizontal, condenser up, and condenser down. While a heat transfer capacity of 4 watts was measured in this experiment, it is possible that heat capacity could be larger. No attempt was made in these experiments to find maximum performance for the two pipes. Any further experimentation should include a method to test these limits. However, given the application in cold weather gear, heat transport capacity as measured in this experiment may well be sufficient to predict success.

VII. SUMMARY

This report describes the testing of two flexible, corrugated stainless steel heat pipes at low temperature for possible application in cold weather clothing. Axial temperature distribution as well as heat transfer capabilities were measured for three different orientations: horizontal, condenser up, condenser down. Results indicated at least four Watts could be effectively transferred by either pipe.

This document reports research undertaken at the US Army Natick Research, Development and Engineering Center and has been assigned No. NATICK/TR-93/005 in the series of reports approved for publication.

VIII. REFERENCES

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